THE EFFICIENT USE OF STEAM

BY

OLIVER LYLE

"In short, I expect almost totally
to prevent waste of steam"

JAMES WATT. Letter to Dr. Lind, 1765

LONDON

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FOREWORD

It is now more than 10 years since Sir Oliver Lyle wrote "The Efficient Use of Steam". In the meantime, over 20,000 copies of the book have been sold. This is a remarkable tribute to the skill of the author in presenting for the first time a mass of technical data and information in the form of a useful and practical guide to all concerned with the use of steam in industry, and a sure indication that such a publication was badly needed.

One of us, as Minister of Fuel and Power, had the pleasure of introducing this book in 1947; now we both welcome the opportunity of commending this new impression and of recording our gratitude to Sir Oliver Lyle for the care and trouble which he has taken to ensure that it has been brought up to date.

We hope that "The Efficient Use of Steam" will have an even wider circulation than before, and that the information and advice which it contains will be applied by users of steam plant throughout the country.

Mills.

E Shinwell

PREFACE TO THE SIXTH IMPRESSION

The opportunity has been taken in this new Impression to bring many of the cost figures up to date, and for some hundreds of minor revisions to be made. Apart from a less dogmatic treatment of the behaviour of superheated steam for heating, there are no important additions, omissions or alterations.

O.L.

INTRODUCTION

> Soon shall thy arm, unconquered steam! afar
> Drag the slow barge, or drive the rapid car;
> Or on wide waving wings expanded bear
> The flying chariot through the field of air.
>
> ERASMUS DARWIN. *The Botanic Garden.* 1792.

STEAM is industry's most wonderful, flexible, adaptable tool. No self-respecting workman presumes to call himself by his craft name until he knows how to use and look after his hammer, chisel, saw, micrometer, megger—or even hod or larry. Yet "Steam Users" misuse and ill-treat steam, the beautiful heating and power tool, because they have never learnt how to use and care for it.

Steam has tremendous advantages over other tools. Once its ways are known, it never slips, never breaks, never turns its edge. Everything, or almost everything, that needs to be known about steam is in the Steam Tables—awe-inspiring columns of figures with unalluring titles.

There is nothing new about steam. Its properties have been known for 100 years. Each generation simply adds another decimal place to these properties.

The new knowledge that the war uncovered is the sad lack in only too many factories, large and small, of the simple principles governing the economic use of steam. Immense pains are often taken to capture an extra 2 or 3 *per cent.* in the boiler house, yet the factory may be using 2 or 3 *times* as much steam as is necessary.

This book is an attempt to gather together, for the benefit of the practical factory steam user, the essential information, which should enable steam to be correctly used. About a third of the book has already appeared in the Fuel Efficiency Committee's Bulletins.

As far as possible, mathematics have been avoided (the most "advanced" mathematics in this book is a pair of simultaneous equations), but there is necessarily a good deal of arithmetic.

Abbreviations have been borrowed from our transatlantic allies; for example, the cumbersome British contraction lb. per sq. in. has been replaced by psi, and Btu has replaced B.Th.U.

The first three chapters are largely theoretical. Some readers may find them a little dry. The author was strongly tempted to put them in an Appendix, but, as all but two of the subsequent chapters are built on the structure of the first three, it was felt necessary to start with the theoretical framework.

Considerable trouble has been taken to make the Index complete. Cross indexing has been done often to the third and fourth generation, but cross indexing is not 100 per cent. complete. By making the Index a section index instead of a page index it was possible to prepare the index unhurriedly while the text was being criticised and corrected.

Where tables are short they are first printed in the text to which they apply. Most of the tables are collected together at the end of the book for easy reference.

The book is intended to be a practical manual; it is not a text book. No attempt is made to describe the design and construction of the more important pieces of steam plant, boilers, engines, turbines, etc., though the behaviour of steam in some of these plants is discussed, often in great detail. Some of the explanations may not be scientifically correct, but the author believes that it is more important to be nearly right and understandable than to be academically accurate and incomprehensible. (The author is insufficiently equipped to be academically accurate.)

The book would never have been possible without the help of many friends whose brains the author has shamelessly picked. In particular the following have given, often unconsciously, of their learning and experience: M. H. Adams, W. L. Badger*, E. H. Blade*, W. L. Boon†, R. Carstairs*, F. M. Chapman, W. Davies*, J. Eisner, Oscar Faber*, M. Francis*, W. O. Goldthorpe†, P. F. Grove†, F. Courtney Harwood†, A. H. Hayes*, J. E. Hobbs†, Philip Lyle*†, E. L. Luly, I. F. G. McVicker†, J. B. M. Mason*†, A. Milnes*†, L. G. Northcroft*†, H. M. Peacock*, J. Philip*, E. G. Ritchie*†, Peter Runge*†, F. C. Sellens†, F. Shakeshaft, C. E. G. Simmons*†, B. Smith†, W. J. Sparkes†, I. M. Stewart*, J. A. Vice†, A. L Webre*. In the case of those whose names bear asterisks, plagiarism has sunk to scissors and paste brush. Those whose names carry daggers have read and criticised text or corrected arithmetic. To all of these the author is indeed grateful. To Ernest Grumell†, Chairman of the Fuel Efficiency Committee, and to Angus Macfarlane*†, Director of Fuel Efficiency, the author owes particular thanks for help and guidance. Much valuable information has also been obtained from manufacturers of repute. Finally the author must thank W. A. Glover for translating some 400 rough sketches into pretty pictures, Phyllis Davies for hundreds of sheets of virtuoso typescript, J. A. C. Hugill† and H. H. Buckley† for reading, correcting and revising the proofs.

Although this book was written for the Fuel Efficiency Committee, although it has been read and criticised by a dozen people and although much of its content has been taken from the work of others, the author alone accepts responsibility for the statements made and the views expressed.

O.L.

CHAPTER 1

WHAT STEAM IS—ITS HEATING PROPERTIES

> For hot, cold, moist and dry, four champions fierce.
> Strive here for mastery.
>
> MILTON. *Paradise Lost.* 1666

1. STEAM. In our factories steam is used for heat and power because it possesses many outstanding qualities, for example :

It has a very high heat content.
It gives up its heat at constant temperature.
It is produced from water which is cheap and plentiful.
It is clean, odourless and tasteless.
Its heat can often be used over and over again.
It can generate power and then be used for heating.
It can be readily distributed and easily controlled.

As far as Steam for Heating and Process is concerned, there are just two fundamental things that govern everything. These are :

1. The boiling point of water decreases with reduced pressure.
2. The Latent Heat (the "heating" heat) of steam increases with reduced pressure.

As far as steam for Power is concerned there are also two basic rules :

(*a*) Use the highest practicable initial pressure and temperature.

(*b*) Use the lowest practicable exhaust or back pressure.

As far as steam for any purpose is concerned there is another rule that is of universal application :

> Never permit steam to expand from one pressure to a lower pressure without getting some useful result from the expansion.

In order to understand Steam it is desirable to have some idea of what is going on inside water and steam under all sorts of conditions.

2. SOLID, LIQUID, VAPOUR. Steam is water vapour. Many things can exist in the three different states—solid, liquid, vapour. We all know that the cooler the substance the more likely it is to be a solid. The hotter it is the more likely it is to be a vapour. What is the mysterious change which affects substances so that a solid "melts" into a liquid and liquid can "dry" or "boil" into a vapour ?

3. MOLECULES. All substances are built up of extremely small particles called "Molecules". These molecules are made up of even smaller particles called "Atoms," which in turn consist of even tinier particles. The essential difference between these various particles is that a molecule is the smallest particle of a substance which possesses all the chemical properties of the substance in bulk ; in other words the qualities that enable us to say : "This is sugar ; that is salt".

Each molecule of ice, water or steam is built up of two atoms of hydrogen and one atom of oxygen. If all the molecules in ice, water or steam are the same, why is ice solid, water liquid and steam vapour ? The answer is that a liquid contains more energy than its solid, and that a vapour contains more energy than its liquid. We might have said that water is warmer than ice, and steam is hotter than water. This answer would not always have been true. Ice and the water it is forming in have the same temperature. The water boiling in the kettle is at the same temperature as the steam coming from the spout.

The energy that molecules or any other substance can possess is of two forms ; kinetic energy—energy of motion, and potential energy—energy of position. An aeroplane in flight has kinetic energy dependent on its speed, and potential energy dependent on its height.

When energy is added to a substance under conditions that do not change the state of the substance, the energy takes the form of increased motion of the molecules and is shown by a rise of temperature. When energy is added under conditions that change the state of the substance, the energy takes the form of change of position of the molecules and is shown as a change of state without any change of temperature.

4. LIQUID. The molecules in a liquid are in constant motion, and their speed of movement is dependent on the temperature. The hotter the liquid the faster do the molecules move. In a liquid the molecules have a great attraction for one another, are very close together, and their range of movement is very limited. Owing to this congestion and to their erratic movement collisions often take place. As a result of multiple collisions some molecules go, for a short time, much faster than their fellows who may have been slowed, or even stopped, as a result of the collision. If such a speed-merchant is travelling upwards at the surface of the liquid it will jump out of the liquid into the air or vapour above. The more heat we add to the liquid the faster go the molecules, the more molecules can escape.

A childish analogy may help to explain how a molecule can reach a speed greatly above the average at the expense of its neighbours as a result of multiple collisions. By squeezing a wet cherry stone between finger and thumb we can impart a very high speed to the stone as a result of its being involved in a low-speed collision between finger and thumb. If we imagine the cherry stone to be one molecule and the finger and thumb each to represent a group of converging molecules, this analogy gives a good picture of molecular acceleration.

5. VAPOUR. In a gas or vapour the molecules have overcome the attraction that held them together as a liquid. They are therefore much further apart. The volume of a pound of steam at atmospheric pressure is sixteen hundred times the volume of a pound of boiling water. In a vapour, as the molecules have overcome their mutual attraction, their movement is quite free and is limited only by collision with each other and with the walls of the container.

The difference between a gas and a vapour is partly one of definition and partly one of behaviour. A vapour is called a vapour when it is at a

temperature at which it can exist as a liquid provided the pressure is sufficient. If a vapour is hotter than the temperature at which it can exist as a liquid, however great the pressure, it is called a gas.

A gas obeys certain well established laws regarding its change of volume with temperature or pressure, while a vapour obeys the rules imperfectly

6. **SURFACE EFFECTS.** At the surface of water two things are happening. A few molecules acquire extra energy from their neighbours and travel so fast relatively that they can overcome their mutual attraction and jump out of the liquid. A hail of air or vapour molecules is raining down on to the water surface. The rain of air molecules knocks most of the ambitious water molecules back into the liquid again, but some lucky ones escape. The air contains some water molecules because it always carries some moisture, so that, although some water molecules are jumping from the water surface some air-borne water molecules are diving back into the liquid.

7. **HEAT IS ENERGY.** Heat is just a form of energy. When heat is added to a substance it is stored in the substance as extra molecular movement. Energy can be used to move masses of molecules in bulk and then we call it mechanical energy, and the result is mechanical motion. The greater the weight to be moved. the greater the energy needed. Similarly, the greater the number of molecules to be heated—that is, to be speeded up—the greater is the energy needed as heat.

8. **DRYING.** Let us consider a puddle. It is a dull sultry day with no wind. A few molecules of water are jumping out of the puddle. Some of them escape into the air over the puddle until the overlying air contains so many water molecules that just as many dive back into the puddle as are jumping out of it. Since the escaping molecules take their energy with them and since they stole their extra energy from their fellows, it follows that the total energy in the puddle is reduced, its temperature is lowered, fewer molecules try to escape, a balance is reached, so the puddle does not dry up any more.

If the sun comes out the puddle is warmed and its molecules move faster so that more can jump out than are diving back, so the puddle starts drying up again. The heat of the sun starts air currents and the air above the puddle which was laden with water molecules, is wafted away from the puddle surface, so that the puddle can dry up still faster.

If foggy air comes along conditions are reversed. Such air carries more moisture than it can hold and some of the water molecules join up as liquid droplets, which will fall into the puddle and it will get bigger. If the puddle is colder than the damp air more water molecules will dive into it than are jumping out, and this will make the puddle bigger still.

The escape of high-speed molecules from a liquid (evaporation or drying) causes a reduction of energy in the remaining liquid molecules. The consequent drop of the liquid temperature causes the evaporation or drying to slow down unless heat energy is added to the liquid to keep up its molecular activity. This effect is very clearly noticed by standing in a breeze after perspiring. The breeze carries away the escaping molecules, their energy is lost to the perspiration which drops in temperature and we feel chilled.

9. WHAT PRESSURE IS. The pressure exerted by a gas or vapour is due to the myriads of impacts of the molecules bombarding the surface enclosing the vapour. If we halve the number of the molecules (without changing their speed by a change of temperature) in a given space, we halve the pressure because we have halved the bombarding missiles. If we add heat to a gas or vapour in a vessel we increase the speed of the molecules and therefore the temperature rises. The faster moving molecules demand more room, so the vapour tends to expand. If we prevent expansion by keeping the vessel closed the faster moving molecules, having the same density as before, must produce a heavier bombarding effect which shows as an increase of pressure.

10. VAPOUR PRESSURE. If the pressure on the surface of a liquid is caused by the rain of air or vapour molecules, it follows that those molecules trying to jump out of the liquid must be exerting a pressure on the air or vapour above the liquid. The pressure at which the escape of molecules from the liquid just balances the overlying pressure is called the Vapour Pressure of the liquid.

11. BOILING. Put cold water into an open vessel under atmospheric pressure. Add heat to the water. The molecules move about faster as more heat is added. This increase of water energy is shown as a rising temperature. More molecules try to jump out of the liquid—its vapour pressure rises. Most of the adventurous water molecules get knocked back into the liquid, so that practically the whole of the added heat energy is retained as increased molecular speed in the liquid. As the addition of heat proceeds we reach a point where the upward bombardment by jumping molecules overcomes the downward bombardment of the overlying air or vapour molecules. That is to say, the liquid vapour pressure overcomes the overlying pressure. The speeding water molecules, having won the battle, can now leave the water freely provided they receive sufficient energy to enable them to overcome their attraction for their comrades. Any additional heat put into the water merely shoots off more vapour molecules. It is impossible to raise the temperature of—that is, to increase the energy in—the water, because this would increase the vapour pressure which cannot rise because it has already overcome the overlying pressure. The particular temperature at which this state occurs is called the "Boiling Point".

12. EFFECT OF INCREASED PRESSURE. If we close up the vessel and pump air into it so that there is a considerable pressure above the liquid there are now more air or vapour molecules bombarding the liquid surface. It will therefore be more difficult for liquid molecules to fly off; more will be knocked back. In order to make a certainty of escape the liquid vapour pressure must be increased to overcome the greater overlying pressure. The temperature must be higher which means that the boiling point must rise with increase of pressure.

13. EFFECT OF REDUCED PRESSURE. If we attach a vacuum pump to the vessel and pump out much of the air there will be fewer air molecules raining down on to the surface. The liquid molecules suffer less interference and can escape more easily. The water can exert the necessary vapour pressure at a lower temperature. So, the boiling point of a liquid falls with reduced pressure.

14. HEAT OF VAPORISATION. When heat is added to liquid which is at its boiling point the fly-away molecules take the added energy with them. They require it to overcome the attraction of their liquid comrades. Once sufficient energy has been added to bring the liquid to boiling point and win the vapour pressure battle, the liquid can gain no more energy. Any attempt to add more energy to the liquid just produces more vaporised molecules. The escaping molecules require two rations of energy to escape. They have their ration of liquid energy and if they pick up a sufficient extra ration to overcome their mutual attractions they escape. The first ration is their "Liquid Heat," "Sensible Heat" or kinetic energy ; the second is the "Heat of Vaporisation," "Latent Heat" or potential energy.

15. CONDENSATION. Suppose that we have a vessel supplied with steam at a constant pressure and that we spray a little cold water into the vessel. The molecules in the cold water are moving slowly so that the vapour pressure is low. The high-speed steam molecules bombarding the water meet little opposition and can plunge joyfully into the water transferring their energy to the water. This increases the speed of the water molecules ; the water temperature rises ; its vapour pressure rises. This process, in which some of the steam molecules are converted into water, will go on until the vapour pressure of the water is equal to the steam pressure—that is to say until the temperature of the water is the same as that of the steam. This process is called "Condensation". Condensation under the conditions just described, where there is an excess supply of steam compared to the supply of water, results in the water being raised to the temperature of the steam—that is to say to the appropriate boiling point.

Suppose that we spray an excess of cold water through the vessel which is being supplied with a relatively small amount of steam. The steam molecules will plunge into the water, but, as the quantity of water is large compared to the amount of steam, the water will not be heated up very much—its vapour pressure will be low. This enables the steam molecules to go on diving into the water until the pressure of the steam has been so reduced as to balance the low vapour pressure of the water. In these circumstances the pressure of the steam is reduced until it is the same as the vapour pressure of the slightly heated water. Condensation with excess water reduces the steam temperature and pressure. It creates a partial vacuum. This is the way a "Jet Condenser" works.

Steam molecules can bombard any substance, solid, liquid, or gaseous, and, if the molecules of the substance are moving slower than the steam molecules, the steam molecules will give up energy in speeding up the slower molecules. The steam molecules, having thus lost energy, will be moving slower and some of them will find that they have not enough energy to overcome their mutual attraction. They will therefore come together again as water. This will occur on the surface of a cool metal and is the explanation of the working of a "Surface Condenser". Both surface condensers and jet condensers were invented by James Watt in 1765.

16. MELTING. A solid differs from a liquid in that the molecules arrange themselves in a form where they exert the greatest possible attraction for one another. In such an arrangement they are locked together and cannot move

independently, their movement is limited to vibration. When a solid is heated its molecules vibrate faster until, at a certain point, some molecules are able to tear themselves away from their fellows and move independently. This is the process called " Melting ". As in vaporisation, the melted molecules take their extra energy with them and prevent further rise of temperature of the solid. The temperature at which this takes place is called the " Melting Point ".

17. HEAT TRANSFER. If heat is added to a metal vessel which contains liquid or vapour, the molecules of the metal vibrate faster. These vibrating molecules strike the fluid molecules blows which increase the speed of the fluid molecules. In this way heat energy is transferred to a liquid or vapour.

If a liquid or vapour inside a vessel is hotter than the metal of the vessel, the faster moving fluid molecules strike the metal molecules and start them vibrating faster. In this way heat energy can be transferred from a fluid to a metal container.

Heat energy must flow from more energetic molecules to those with less energy. Substances therefore always tend to equalise in temperature if they are in contact.

18. CONDUCTION, CONVECTION, RADIATION. When a solid is heated the vibrations of the heated molecules are transferred to their neighbours by a kind of shoulder-jostling process. In a liquid or vapour the molecules transfer added heat to their neighbours by collisions. These forms of heat transfer by impact are called " Conduction ". Materials vary greatly in the speed with which they will transfer heat by conduction. Solid metals are excellent conductors ; non-metals are poor conductors. Liquids are generally worse conductors than solids, and gases or vapours are the worst conductors of all.

There is a method by which liquids and gases can receive heat quite quickly in spite of being poor conductors. Nearly all substances expand or get bigger when heated, because their more energetic molecules demand more elbow-room. When a liquid or vapour is in contact with a hotter metal, the fluid molecules alongside the metal get speeded up almost instantly although they transfer the movement to their neighbours slowly. These more active molecules demand more room, the hot layer expands and is therefore lighter than the remaining fluid. The hot layer rises and cooler fluid flows in to replace it and is in turn brought into contact with the hot surface. This process is called " Convection " and is just a combination of short-distance conduction and movement.

The molecules of a hot substance, in addition to transferring heat energy by molecular impacts, also have the property of transmitting energy as radiations across empty space. The sun's energy reaches the earth by this means. From a steam user's point of view radiation is of no importance as steam receives and gives up its heat by conduction and convection, although the boiler *metal* may receive heat by radiation and the steam *pipe* may give up heat by radiation.

19. LAGGING. Gases are bad heat conductors and they absorb and transfer heat principally by convection. If a gas can be kept in contact with a hot

surface in such a way that convection cannot take place, heat transfer will be slow. Lagging materials, diatomite, cork, asbestos, hair, etc., entangle air and impede its convection movement. That is why they prevent a surface losing much heat. Some cheap lagging materials act more as non-conductors than as non-convectors.

20. SATURATED STEAM. When vapour is at a temperature corresponding to the liquid boiling point appropriate to its pressure it is said to be "Saturated". When no liquid is present at that temperature it is called "Dry Saturated".

21. SUPERHEATED STEAM. As long as steam is in the presence of water it is impossible to raise the temperature of the steam above the appropriate boiling point. Any attempt to add more heat energy will simply vaporise more water. If heat is added to steam that is not in contact with water the steam molecules will become still more energetic and will jostle each other more rapidly. The temperature will rise; the steam is "Superheated". Superheated steam occupies more space than it did in the saturated state because the more active molecules drive each other farther apart. As explained in Section 9, if the steam is confined in a closed vessel the extra molecular energy added in superheating will result in an increase of pressure.

22. THE COOLING OF STEAM. Saturated steam has only just sufficient energy to keep it in the vapour state. When it gives up heat to a cooler body some of it condenses and the condensing molecules give up all the potential energy that they took on as their extra ration when they overcame their liquid attraction. If superheated steam is cooled, the superheat must first be given up. Were some molecules to become liquid while others were superheated, the more active superheated molecules would goad the lazy liquid molecules back into vaporous activity. So that superheated steam must first part with its superheat before it can condense.

23. WET STEAM. If saturated steam is in rapid turbulent motion through a pipe on whose surface some of it is condensing, some of the water may be swept along with the steam which then becomes a mixture of true vapour and liquid droplets. Steam in this condition is called "Wet Steam". There is another way in which steam can become wet after expanding in an engine or turbine. In this type of wet steam the moisture is much more finely divided and the steam is an extremely fine mist.

Steam will also be wet if water drops or foam are carried out of a steam boiler with the steam.

24. SUBMERGED BOILING. If heat energy is added very fast to a liquid the added energy does not spread instantly throughout the liquid. Convection and conduction both take time. A group of molecules may therefore receive sufficient energy to exert a vapour pressure high enough to overcome the pressure acting on the liquid. They will therefore escape in the form of a "bubble". If the water is relatively cool these bubbles will condense as they rise and as they collapse they will make small sounds. The singing of a kettle

is simply the sound made by streams of tiny condensing steam bubbles. If the liquid is at boiling point the bubbles will get bigger as they rise until they break the surface with quite a different noise.

25. PRESSURE MEASUREMENT. We measure pressure as pounds per square inch. Assume we have a vessel 28 in. high with a cross section of 5 in. × 2 in. Fill this vessel within a quarter of an inch of the top with water. It will hold a gallon, or 10 pounds. The base of the vessel has an area of 5 in. × 2 in. or 10 sq. in. So we have 10 pounds pressing on 10 sq. in., or 1 pound per square inch (lb./sq. in., or psi). We see from this that a column of water nearly 28 in. high exerts a pressure of 1 psi.

26. ATMOSPHERIC PRESSURE. All the molecules in the atmosphere are being drawn towards the earth by its gravitational pull, and the air near the earth's surface is being pressed upon or bombarded by the molecules in the upper air. It follows that the air nearer the earth is at a greater pressure and is denser than the air in the upper regions. The upper air gets thinner farther from the earth until, about 500 miles up, we can say there is no more air. The combined effect of all these miles of air is that the air molecules at the earth's surface are bombarding everything with a force of nearly 15 psi (actually 14·696 psi at sea level in normal good weather).

27. PRESSURE GAUGES. When, at Christmas or other beano, we blow out the rolled-up flat tube of a paper tongue toy, we measure the pressure exerted by our lungs by the length which the paper tongue extends. The ordinary dial pressure gauge works exactly the same way. The steam or other pressure is led into the inside of a curved flattened tube which tends to straighten the greater the pressure. The pressure of the atmosphere is acting on the outside of the tube, tending to keep it flat and bent ; so that the pressure gauge really measures the amount by which the pressure being measured exceeds the atmospheric pressure. This pressure is called the "Gauge Pressure ". The real or "Absolute Pressure" is the gauge pressure plus the atmospheric pressure. So a gauge pressure of 15·3 psi.g. is an absolute pressure of 30 psi.a. and a gauge pressure of 0 psi.g. is an absolute pressure of 14·7 psi.a.

28. MERCURY GAUGES. A column of water nearly 28 in. high exerts a pressure of 1 psi. Mercury is about 14 times as heavy as water, so that a column of mercury about 2 in. high will exert a pressure of 1 psi. To exert a pressure equal to that of the atmosphere—14·7 psi—will require a mercury column about 30 in. high. The ordinary dial pressure gauge, working by means of the curved flattened tube, is not very satisfactory for measuring pressures below atmospheric. Such low pressures are better measured with a mercury column, made from a glass U-tube. If the U-tube has one end open and the other end connected to the vessel whose pressure is to be measured, the difference in heights of mercury in the two arms shows the difference above or below atmospheric pressure.

If one arm of the U-tube is closed, and all the air is removed from the closed end, the pressure shown by the difference in heights of the two columns is the absolute pressure in "Inches of Mercury" (In.Hg.). If the pressure in the

vessel is above atmospheric pressure the vessel is said to be " Under Pressure ". If the pressure is below atmospheric pressure the vessel is said to be " Under Vacuum ". If we are measuring vacuum the mercury column may show 2 in. of absolute mercury pressure. In industry this is usually called 28 in. of vacuum—that is to say, the pressure is 28 in. less than atmospheric pressure. The mercury gauge is not convenient for measuring high pressures owing to the great height of mercury that would be needed.

29. THERMOMETERS. Nearly every substance expands as it gets hotter, but the amount of expansion for a given temperature rise differs greatly between substances. Solids expand less than liquids ; liquids expand less than gases. This property is used in most low-temperature thermometers. When a mercury in glass thermometer is heated the mercury expands more than the glass, so the mercury pushes its way up the tube and registers temperature by its height on a suitable scale.

30. TEMPERATURE SCALES — FAHRENHEIT. There are several thermometer scales. The scale in most general use in Britain is the " Fahrenheit " scale, and this is the scale used in this book. Fahrenheit was a German physicist who devised the mercury thermometer about 1720. His scale appears to us to be clumsy, having 32° as the melting point of ice and 212° as the boiling point of water ; but, with the means at his disposal, Fahrenheit is not to be blamed. He based his zero on the greatest cold he could get from a freezing mixture of snow and salt. This temperature he called 0. As his hot point he took his body temperature which he appears to have called 12 degrees. He then seems to have divided his 12 main degrees into eighths like an ordinary foot-rule. Using this scale, with a body temperature of 96 small (eighth) degrees he gave the melting point of ice as 32°. At that time the boiling point of water was not known and was later found to be 212 small Fahrenheit degrees. Subsequent measurements, basing the Fahrenheit scale on a water freezing point of 32° and a water boiling point of 212°, gave a body temperature of 98·4°.

31. CENTIGRADE. The rise of temperature, on the Fahrenheit scale, from water freezing point to water boiling point is $212 - 32 = 180°$. About 1740 Celsius suggested that a more rational scale would be to use water boiling point as zero and water freezing point as 100. Linnaeus persuaded Celsius to reverse his scale and the " Centigrade " scale with water freezing point at 0° and water boiling point at 100° was made public in 1745. This scale is used for all laboratory and scientific purposes and is used in an increasing number of industries.

32. HEAT MEASUREMENT. Suppose we add equal quantities of heat to four vessels each containing an equal weight of one of four different liquids : water, alcohol, " Pyrene " and mercury. If the water is raised in temperature by 10° F., the alcohol will rise by 16° F., the " Pyrene " by 50° F. and the mercury by 303° F. Clearly substances vary in their heat capacity and some rise in temperature much more for a given heat addition than others. So we cannot use temperature rise as a measure of heat addition without specifying the substance being heated. The standard unit of heat is the quantity of heat

needed to raise the temperature of one pound of water by 1° F. at room temperature. This amount of heat-energy is called a "British Thermal Unit" or B.Th.U. or B.u.

If the Centigrade scale is being used, the amount of heat needed to raise one pound of water by 1° C. is the "Centigrade Heat Unit" or C.H.U. or "Pound-Calorie". 1 C.H.U. = 1·8 Btu.

Now the unit of heat, the Btu, cannot be measured directly on any kind of meter. If we had 10 Btu this could either warm 1 pound of water 10 degrees, or 10 pounds of water 1 degree, or 100 pounds of water 1/10 degree. Consequently in order to measure Heat Flow or Heat Addition we must measure a change of temperature in a measured amount of material, and the amount of heat that the material can carry, absorb or give up must be known.

33. SPECIFIC HEAT. The amount of heat that a substance can hold relative to water is called the "Specific Heat" and is the amount of heat in Heat Units needed to raise the temperature of one pound of the substance by one degree of temperature. The specific heats of the liquids just mentioned are :—

Water	1·00
Alcohol	·63
Pyrene	·20
Mercury	·033

The specific heats are not constant but vary with temperature and pressure. The specific heat of steam at low pressure and moderate temperature is about half that of water. The specific heat is the same whether measured as Btu per °F. or C.H.U. per °C.

34. THE STEAM TABLES. We are now in a position to understand everything in the Steam Table except Entropy, and Gibbs' Function. Entropy is a somewhat obscure quality and refers to the availability of power energy in steam. It is generally only used by power engineers, but its proper understanding is of great help in all steam problems. It will be dealt with in Chapter 2. Gibbs' Function is simply an arithmetical short cut and its use will also be described in the next Chapter. As far as heating is concerned, neither entropy nor Gibbs' Function need be used. The remainder of this Chapter will be devoted to a study of the steam tables and the uses that can be made of the information they provide for solving all heating problems.

Steam Tables are printed on pages 821-836. Table I gives the properties of Saturated Steam; Table II the properties of Superheated Steam. The diagrams, Figs. 1, 2 and 3, are drawn from the information given in Table I. Turn to the steam tables and study them. Get quite familiar with them. They will prove good friends.

35. PRESSURE AND BOILING POINT. The first three columns of the Saturated Steam Table show the water boiling points corresponding to various gauge and absolute pressures. It has already been explained (Sections 10–12) that high pressure calls for a high vapour pressure to permit boiling and this can only be obtained by a higher temperature. Low pressure only needs a low

vapour pressure to win the boiling battle, so that the temperature can be low. Fig. 1 is a curve, drawn from the Steam Table, showing the relation between boiling temperature and pressure over the low-pressure range.

There are six ways in which advantage can be taken of this useful relation :—

(1) The low temperature evaporation of delicate products.
(2) The use of low-pressure steam for evaporation.
(3) The multiple effect evaporator.
(4) The steam accumulator.
(5) Flash cooling.
(6) Lowering boiler pressures.

Some of these can be dealt with at once, but others must wait till we have delved deeper into the Steam Tables.

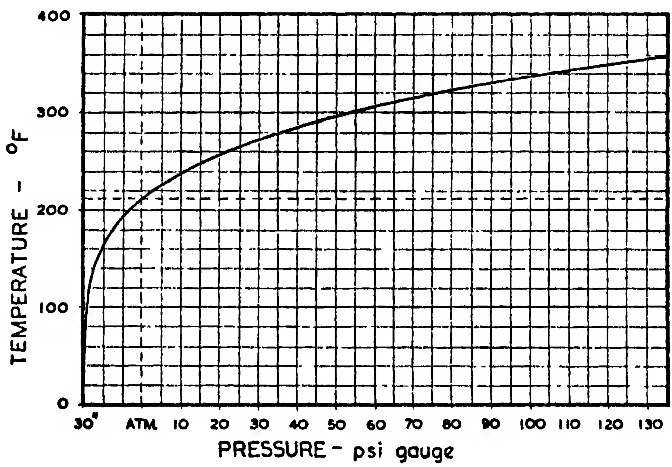

FIG. I. VARIATION OF WATER BOILING POINT WITH PRESSURE

36. LOW TEMPERATURE EVAPORATION. If, for some reason, it is necessary to boil a watery liquor at a low temperature, the Steam Table and Fig. 1 show that this can be done by boiling under vacuum. For example, milk consists largely of water and it is damaged by high temperatures. By boiling milk in a closed evaporator, whose outlet is connected to a condenser and vacuum pump, it is possible to concentrate it at about 25 in. vacuum, when the temperature can be kept below 140° F.—to prevent destruction of vitamins.

37. USE OF LOW PRESSURE STEAM. Suppose we have process washings to be evaporated and suppose there is a lavish supply of low-pressure steam in the form of, say, exhaust from pumps. Assume that hitherto the washings have been concentrated under atmospheric pressure using 20 psi.g. steam in the evaporator heating surface. The 20 psi steam must be condensing and giving up its latent heat to the heating surface at its saturation temperature which the Steam Table tells us is 259° F. The washings will be boiling at 212° F. There is therefore a 47° F. temperature difference across the evaporator heating surface. Suppose that, instead of using 20 psi steam we wish to use pump exhaust at atmospheric pressure with a temperature of 212° F. Low temperature

heat does not pass so readily through a heating surface as high temperature heat because the molecules do not vibrate so energetically at low temperature; the shoulder-jostling process is not so vigorous. We must therefore arrange for a larger temperature difference across the heating surface. If we decide on a 60° F. difference, the washings must be made to boil at $212 - 60 = 152°$ F. The Steam Table tells us that we must, with the aid of condenser and vacuum pump, do the evaporation under a vacuum of about 22 in., when the boiling point of the water or washings will be 152° F.

38. MULTIPLE EFFECT EVAPORATION. Instead of using pump exhaust we might have used vapour off another evaporator. Such an arrangement can be made to use the heat in steam over and over again, each vessel receiving steam boiled off its predecessor which of course must be working at a higher pressure. This is called "Multiple Effect Evaporation" and is dealt with in Chapter 17.

39. LOWERED BOILER PRESSURES. The lower the steam pressure in a boiler the lower is the boiling temperature of the water. (It will be seen later that for heating and process purposes steam should be always used at the lowest possible pressure.) The lower the water boiling temperature the more easily will the heat from the furnace gases pass into the boiler. It will also be possible to extract more heat from the flue gases. The improvement will be very small, but it will be there and will be greater where no economiser is fitted.

Care must be taken in reducing the working pressure of a water-tube boiler as the tubes may not be large enough to pass the greater volume of steam which a reduced pressure entails. Carry-over or priming might occur. The water circulation in the tubes may be so slowed down by the congestion due to the large volume of steam that the tubes may get overheated and fail. With any type of boiler a reduction of pressure may not be possible owing to steam pipes being too small. The volume columns in the Steam Table show that steam at 40 psi.g. has double the volume of steam at 100 psi.g. and that steam at 100 psi.g. has double the volume of steam at 220 psi.g. Table III shows the amount of steam that can be expected to flow through pipes under various conditions. Any reduction of pressure that can be made should be made. There are other advantages in the boiler house apart from improved heat transfer. The lower the pressure the less the power needed by the feed pump. The lower the pressure the less total heat is needed for each pound of steam. (This is only true below about 450 psi; above this pressure the reverse occurs.)

40. HEAT. The fourth, fifth, sixth and seventh columns in the Saturated Steam Table are the heat or energy values. The first heat column shows the heat needed to bring one pound of water from the freezing point to the boiling point corresponding to any particular pressure. In Section 12 it has been explained that higher pressure calls for a higher boiling temperature. To reach this higher temperature naturally requires more heat. This water-heating heat is the "Sensible Heat" of the liquid.

The next column shows the amount of heat needed to vaporise completely one pound of water which is already at boiling point. This heat is the heat-energy necessary to enable escaping molecules to overcome their mutual

attraction and exist in a more open formation. If it is added to a liquid at boiling point it forces the molecules apart and makes them overcome their attraction for one another. It is called " Heat of Vaporisation " or " Latent Heat ". This heat energy is the potential energy that must be given up during condensation.

TABLE III. FLOW OF STEAM THROUGH PIPES

Approximate Weight in Pounds of Dry Saturated Steam per Minute that will flow through 100 ft. of various Sizes of Piping with a loss of One Pound per Square Inch of Pressure :

PRESSURE	DIAMETER OF PIPE IN INCHES											
PSI.G.	¾"	1"	1¼"	2"	2½"	3"	4"	5"	6"	8"	10"	12"
5	·53	1·17	3·50	7·4	13·0	20·7	42·7	75·0	118	247	430	717
15	·67	1·42	4·25	9·0	15·7	25·0	51·8	91·3	143	300	523	875
30	·80	1·67	5·16	10·8	19·1	30·3	62·7	110	173	362	632	1055
45	92	1·97	5·83	12·4	21·7	34·7	70·0	126	198	413	722	1203
60	1·00	2·17	6·50	13·8	24·2	38·7	79·7	140	221	460	803	1340
80	1·13	2·42	7·16	15·5	27·2	43·3	89·2	157	247	515	900	1500
100	1·25	2·67	8·00	16·9	29·6	47·3	96·3	172	271	565	983	1645
120	1·37	2·92	8·85	18·6	32·7	52·0	108	188	297	620	1083	1800
150	1·48	3·17	9·50	20·0	35·3	56·0	116	204	322	672	1172	1950
200	1·67	3·67	10·83	22·9	40·3	64·2	133	233	367	767	1333	2230
250	1·87	4·00	12·00	25·3	44·6	71·0	147	258	407	847	1475	2470
300	2·00	4·33	13·00	27·2	47·6	76·0	157	275	433	905	1580	2630

(See Sections 171 to 174)

The third heat column shows the sum of the first two—namely, the " Total Heat " needed to raise 1 lb. water from freezing point to boiling point and completely vaporise it into dry saturated steam. It is the total heat in steam because, as explained in Section 14, vaporising water molecules take both their energy rations with them.

The last heat column headed GIBBS is for use in the arithmetic of the use of steam for power purposes and is described in Chapter 2.

41. SENSIBLE OR LIQUID HEAT. If the specific heat of water were always 1·0 the figure for liquid heat would always be equal to the boiling temperature (or saturation temperature as it is often called) minus 32. But the specific heat rises with temperature and the simple relation does not hold good at high pressure. The change of specific heat with temperature can be seen by comparing boiling temperature with liquid heats :—

Pressure psi.a.	*Temp. °F.*	*Temp. minus 32*	*Liquid Heat*
1·0	101·7	69·7	69·7
10·0	193·2	161·2	161·2
100·0	327·8	295·8	298·4
1000·0	544·6	512·6	542·4

It will be seen that changes in the specific heat of water can be ignored at ordinary low process pressures.

Mistakes are frequent in water heat calculations due to failure to add or subtract 32. This is the chief disadvantage of the Fahrenheit scale. Such errors cannot occur with the Centigrade scale. Water at 69·7° C. contains 69·7 C.H.U.

This heat that must be added to a liquid to raise its temperature up to the boiling point can be detected by our senses as causing a rise of temperature. If we put our hand into water that is being heated our sense of warmth tells us it is getting warmer. If we put a thermometer into it our sense of sight tells us that the temperature is rising. So this liquid heat whose addition can be detected by our senses is called " Sensible Heat ".

42. LATENT HEAT. The heat energy needed to overcome the liquid molecular attraction and cause water molecules to escape as vapour does not show itself as a rise of temperature. Our senses only perceive the change of state from liquid to vapour. Only our memory tells us that energy had to be added. This heat of vaporisation lies hidden in the steam as potential energy and is therefore called " Latent Heat ".

We see from the Table that as the pressure rises the amount of latent heat gets less, until, at a very high pressure it disappears altogether. At high pressures the vapour molecules are more tightly squashed together than at low pressures. If the pressure is raised sufficiently there must come a point where the volume occupied by the vapour is the same as that occupied by the liquid. This point is called the " Critical Pressure " and is 3,208 psi.a. for water. If at this critical pressure water and steam occupy the same space the molecules must be equally closely packed in both states. To effect vaporisation clearly does not call for any extra energy ration because the vapour molecules are no farther apart than liquid molecules. At the critical pressure, therefore, as soon as the " Critical Temperature " of 705·6° F. is reached, the water changes instantly, quietly and without " boiling " into vapour. We have one pound of water at 705° F. one instant, and one pound of steam at 706° F. the next instant. There is no constant temperature boiling point while heat is being absorbed to effect evaporation.

As the pressure increases therefore the amount of energy called for by escaping molecules must be less because their possible freedom, or distance apart, gets less the more tightly they are packed.

At reduced pressure the difference between the space occupied by vapour and water molecules increases so that more energy is required to bridge the gap between liquid and vapour. Inspection of the volume columns of the Steam Table tells us that steam at atmospheric pressure occupies 1,600 times* the volume of water, whereas steam at 25 in. vacuum occupies nearly 9,000 times the volume of water. Clearly a molecule escaping at 25 in. vacuum will require more energy to justify its escape into its larger " lebensraum ".

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* In 1764 James Watt measured the expansion of water into steam at atmospheric pressure and obtained an 1800-fold volume increase. In the same year he measured the latent heat of steam at atmospheric pressure and found that it was " exactly six times the heat needed to heat well water to 212°F." If Watt's well water was at 50°F. his measurement of latent heat gave a figure of 972 Btu/lb. The latest figure, obtained 175 years later, is 971 Btu/lb.

There is therefore a simple explanation for the increase of latent heat with reduced pressure and for a lower latent heat at high pressure.

43. THE HEATING VALUE OF LOW PRESSURE STEAM. What can be done to make use of the fact that the latent heat of steam is greater the lower the pressure ? When steam is used inside a heating surface, coil, pipe or jacket, the steam condenses and gives up its latent heat. All the liquid or sensible heat remains in the condensate which is removed by the trap. The lower the pressure the greater the latent heat and the less the sensible heat. It follows that the greatest amount of heat can be obtained from the condensation of heating steam by using the lowest possible pressure. Where the heating surface drains slowly the condensate will sometimes give up some of its sensible heat. This is usually unintentional and it is generally due to bad design, insufficient fall, faulty trapping, etc., though occasionally a thermostatic trap is used to keep the heating surface partly waterlogged so as to use the sensible heat of the condensate.

It will be seen from the Steam Table that steam at 20 psi.a. contains 5 per cent. more latent heat than steam at 65 psi.a. If therefore a plant could be adapted to use 20 psi steam instead of steam at 65 psi, we should use 5 per cent. less steam. As 20 psi.a. steam contains 2½ per cent. less total heat than 65 psi.a. steam 2½ per cent. of coal will be saved in the boilers, giving a total saving of 7½ per cent.

Fig. 2 shows the proportions of latent heat and sensible heat in low pressure steam. The latent heat is shown below the sensible heat for diagrammatic clarity. Fig. 2 shows how much good heat is retained in the condensate at higher pressures. As a general rule sensible heat is not used in a heating surface because it requires special measures, more plant, etc.

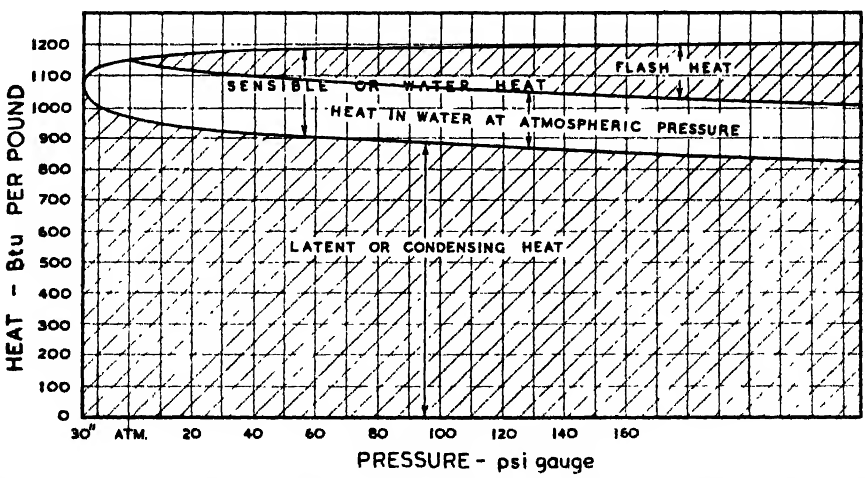

FIG. 2. LATENT HEAT, SENSIBLE HEAT AND FLASH HEAT AT VARIOUS PRESSURES

There may be many cases where advantage cannot be taken of this useful steam property, because the heating surface in the plant may be too small to

give a proper heat transfer rate. Some processes require a certain minimum temperature, the vulcanisation of rubber for example, and a sufficient pressure must be used to give such special temperatures.

One disadvantage of very low pressures is the need for larger steam mains. The volume columns in the table show that steam at 20 psi.a. has more than three times the volume of steam at 65 psi.a. Table III shows that the steam flow at 20 psi.a. will be about half that at 65 psi.a. Such a reduction in pressure may therefore not be possible ; but any lowering of the pressure is a step in the right direction.

44. FLASH. Suppose there is a vessel heated by a steam coil taking steam at 50 psi.a. and suppose the condensate from the trap is piped to a condensate tank open to the atmosphere.

TABLE V. FLASH FOR 1 LB. CONDENSATE COLLECTED AT 212° F.

PRESSURE PSI.G.	WEIGHT OF INPUT STEAM DRY SATURATED	LATENT HEAT GIVEN UP IN PLANT	WEIGHT OF FLASH	HEAT IN FLASH
	LB.	Btu.	LB.	Btu.
5	1·0161	976·2	·0161	18·5
10	1·0294	981·0	·0294	33·9
15	1·0410	984·8	·0410	47·2
20	1·0512	988·2	·0512	58·9
25	1·0607	991·3	·0607	69·8
30	1·0692	994·0	·0692	79·6
35	1·0772	996·3	·0772	88·9
40	1·0848	998·5	·0848	97·6
45	1·0918	1,000·3	·0918	105·7
50	1·0987	1,002·2	·0987	113·5
60	1·1109	1,005·7	·1109	127·6
70	1·1226	1,009·0	·1226	141·1
80	1·1334	1,011·8	·1334	153·5
90	1·1436	1,014·4	·1436	165·3
100	1·1530	1,016·5	·1530	176·1

Table I tells us that each pound of condensate (water) at 50 psi.a. contains 250 Btu.

Each pound of water at atmospheric pressure and boiling temperature contains 180 Btu

The high pressure water (or condensate) therefore contains 70 Btu more than it can hold at atmospheric pressure. The molecules in the hot condensate from the trap are moving at such a speed as to exert a vapour pressure of 50 psi.a., whereas in the condensate tank this is only opposed by a pressure of 14·7 psi.a. There will therefore be a great flash of fly-away molecules until the water energy has been so reduced that it can exert a vapour pressure no more than atmospheric pressure. The heat energy that each pound of condensate must get rid of in this way is 70 Btu.

The latent heat of steam at atmospheric pressure is 971 Btu.

It follows that $\frac{70}{971}$ = ·072 lb. of steam will flash off each pound of condensate, i.e. 7·2 per cent.

In addition to the 70 Btu of latent heat this flash steam will carry away 180 × ·072 = 13 Btu of sensible heat.

Flash steam can be piped to an evaporator, calorifier or vat, or can be used for heating in a contact heater.

Fig. 2 shows the amount of heat that will be liberated as flash if condensate is reduced to atmospheric pressure. It shows clearly how the latent heat rises and the flash drops with lower pressures. Flash is good valuable heat and should not be wasted, but it is a nuisance and needs piping and plant for its collection. If steam can be used for heating at or below atmospheric pressure, not only is there a bigger supply of latent heat than with pressure-steam, but there will be no flash to waste or collect.

Fig. 3 shows the percentage of flash steam produced when the pressure on condensate is reduced. Tables IV and V at the end of the book give figures for flash set out in different ways.

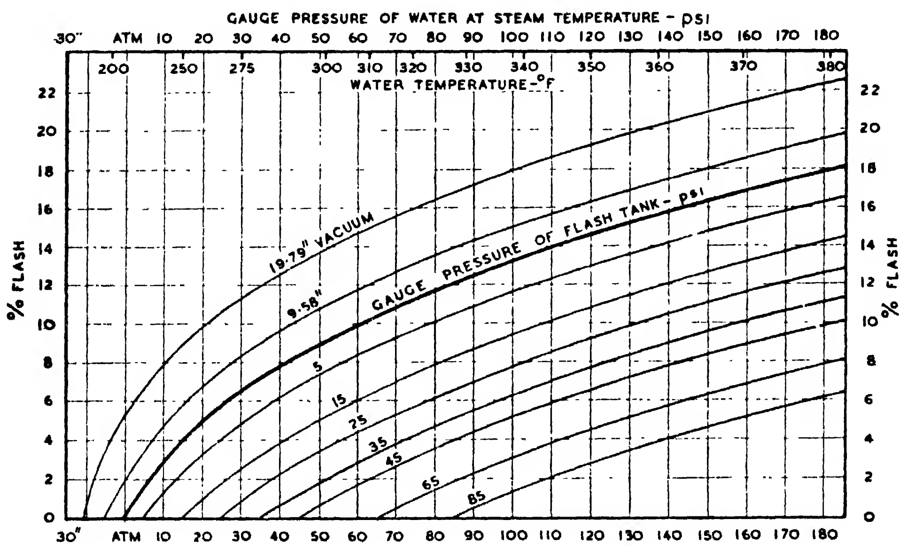

FIG. 3. AMOUNT OF FLASH STEAM FROM WATER AT SATURATION TEMPERATURES

Suppose we have condensate at 110 psi.g. leaving a trap and discharging at atmospheric pressure into a flash tank. How much flash steam will be liberated? In Fig. 3 the intersection of the 110 psi line and the atmospheric flash tank pressure curve (heavy line) occurs just below 14 per cent. flash. Table IV gives the flash under these conditions as 13·94 per cent.

Sometimes the hot condensate can be returned under pressure to the boilers, when there is of course no loss by flash and high pressure steam can then be used in heating surfaces with the advantage of a higher rate of heat transfer. Where arrangements can be made to collect and use the flash steam there is also no loss in using high pressure steam. However both these arrangements require more and special plant and their application is limited. High pressures mean high temperatures so that the heat losses will always be higher.

The use of high pressure steam can only be justified if the process temperatures or the high cost of the plant call for it. High pressure steam should, if possible, first pass through an engine and generate power, the exhaust steam then being used for heating.

The collection of flash steam, the arrangement of flash tanks, and the utilisation of flash steam are dealt with in detail in Chapter 14.

45. STEAM ACCUMULATORS. Suppose we have a steady demand for heating steam and we have an engine working intermittently like a winding engine or rolling mill engine. We can store the exhaust steam from the engine in an accumulator by blowing the exhaust steam into water that is at a lower temperature than the saturation temperature of the blown-in steam. The steam will condense in the water, raise its temperature and the pressure will rise. By reducing the pressure on the water in the accumulator the stored energy will flash off as steam at a lower pressure. Suppose the accumulator contains 10,000 gallons of water at 240° F. corresponding to a pressure of 10 psi.g., and suppose that the engine exhausts at 35 psi.g., 10 per cent. wet, at the rate of 30,000 lb./hour more than the process demands for two minutes and then stops for a few minutes.

The water in the accumulator is 100,000 pounds at 240° F. containing 208 Btu per pound.

The total heat in the accumulator is 20,800,000 Btu.

The surplus engine exhaust is $\dfrac{30{,}000 \times 2}{60} = 1{,}000$ lb.

This steam is 10 per cent. wet so that there will be 100×250 (the water heat at 35 psi.g.) plus $900 \times 1{,}174$ (the total heat of 35 psi.g. steam) $=$ 1,081,600 Btu in the exhaust.

If this exhaust steam is bubbled into the accumulator water it will add its weight and its energy to that already there.

There will be 101,000 pounds of water containing 21,881,600 Btu.

Each pound will contain 216·6 Btu corresponding to a pressure of 14 psi.g.

If the accumulator feeds a 10 psi.g. main each pound of water can give up 8·6 Btu or a total heat of 868,600 Btu.

The latent heat of 10 psi.g. steam is 952 Btu so that the amount of steam flashed off the accumulator will be 912 lb.

Steam accumulators are large, and, where high pressures are wanted, costly. A shell-type boiler, Lancashire, Cornish, Economic, Vertical, etc., acts as an accumulator if the pressure is allowed to fall when a rising load occurs, and to rise with a reduced load. An average Lancashire boiler working at

100 psi will store 320 pounds of steam if the pressure is allowed to rise to 110 psi. This corresponds to about three minutes normal boiler output. If, when a sudden demand for steam occurs, the pressure is allowed to drop from 110 psi to 90 psi, the boiler will give up an extra 630 lb. of steam in addition to its normal output. This is equivalent to over five minutes ordinary steaming. Spare Lancashire boilers can sometimes be adapted to act as accumulators with comparatively small modifications. (In 1943 a battery of four old Lancashire boilers was rigged up to act as an accumulator in a London factory. With a pressure range between 10 psi.g. and 40 psi.g. they had a combined storage of 24,000 lb. of steam.)

Accumulators are discussed at length in Chapter 16.

46. FLASH COOLING. Suppose we have a dilute process liquid at 200° F. and suppose that the next process requires a temperature of 150° F. There may be a heat exchanger available, there may not. Heat exchangers are costly and they need a temperature drop across the heating (or cooling) surface. Another way of cooling is to spray the liquid into an empty vessel connected to a condenser and vacuum pump.

If a 22½-in. vacuum is maintained in the vessel, the steam table tells us that the liquid must boil at 150° F.

It will therefore flash off its surplus heat and reduce its temperature from 200° F. to 150° F.

The steam table tells us that each pound of liquor must flash off 50 Btu.

The latent heat of steam at 22½ in. vac. is 1,008 Btu, so that $\frac{50}{1{,}008} = \cdot 05$ lb. of flash steam will be produced from each pound of liquor.

This effects a 5 per cent. concentration of the liquor and, if the condenser takes the form of a contact heater, process water can be heated with the flash vapour, or space heating can be done. (*See* Chapter 14.)

47. SUPERHEATED STEAM. The superheat table shows the volume, total heat and entropy for superheated steam over a wide range of pressure and superheat. The values at Saturation or boiling temperature are included for comparison.

48. WIREDRAWING. If the pressure on hot water is reduced, we have seen that any surplus heat is given up as self-evaporation or flash. What happens if the pressure on dry saturated steam is reduced ? Suppose we allow dry saturated steam at 150 psi.g. to pass through a reducing valve into a low pressure main at 50 psi.g.

Saturated steam at 150 psi.g. contains 1,196·6 Btu/lb. of total heat.

In expanding through the reducing valve the steam does no work so it still holds those 1,196·6 Btu.

If we look up 50 psi.g. steam in the superheat table we see that 1,196·6 Btu is the total heat of steam at 328° F.

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48. WIREDRAWING. If the pressure on hot water is reduced, we have seen that any surplus heat is given up as self-evaporation or flash. What happens if the pressure on dry saturated steam is reduced ? Suppose we allow dry saturated steam at 150 psi.g. to pass through a reducing valve into a low pressure main at 50 psi.g.

Saturated steam at 150 psi.g. contains 1,196·6 Btu/lb. of total heat.

In expanding through the reducing valve the steam does no work so it still holds those 1,196·6 Btu.

If we look up 50 psi.g. steam in the superheat table we see that 1,196·6 Btu is the total heat of steam at 328° F.

The temperature of 50 psi.g. saturated steam is 298° F., so that the reducing, expansion, " throttling " or " wiredrawing " has added 30° F. of superheat although the actual temperature has fallen from 366° F. to 328° F.

When steam, particularly wet steam, expands through a valve that is cracked open, it is very apt to score the valve seat with grooves that look as if a wire had been drawn through the valve—hence the name " wiredrawing " for the expansion of steam that does no work. Another explanation of the word 'wiredrawing' is that steam passing through an orifice is like a wire being drawn through a die.

This superheating by expansion only occurs to saturated steam below 450 psi. At higher pressures there are other effects because the reduction in latent heat becomes more rapid than the increase in liquid heat. For example, saturated steam at 750 psi blown through a reducing valve gets wetter down to 450 psi and then gets drier until at about 240 psi it is again dry saturated steam. If it is still allowed to expand it superheats itself.

TABLE VI. SUPERHEAT DUE TO WIREDRAWING

INITIAL SATURATED STEAM PRESSURE PSI.G	APPROXIMATE SUPERHEAT IN °F. WHEN INITIAL STEAM BLOWN DOWN TO PRESSURES PSI.G :—														
	2	4	6	8	10	15	20	25	30	35	40	45	50	75	100
5	6	2													
10	14	10	6	3											
15	22	18	14	11	8										
20	28	24	20	17	14	6									
25	34	30	26	23	20	12	6								
30	38	34	30	27	24	16	10	4							
35	42	38	34	31	28	20	14	8	4						
40	46	42	38	35	32	24	18	12	8	4					
45	48	44	40	37	34	26	20	14	10	6	2				
50	52	48	44	41	38	30	24	18	14	10	6	4			
60	58	54	50	47	44	36	30	24	20	16	12	10	6		
70	62	58	54	51	48	40	34	28	24	20	16	14	10		
80	66	62	58	57	52	44	38	32	28	24	20	18	14	2	
90	70	66	62	59	56	48	42	36	32	28	24	22	18	6	
100	74	70	66	63	60	52	46	40	36	32	28	26	22	10	
120	78	74	70	67	64	56	50	44	40	36	32	30	26	14	4
140	84	80	76	73	70	62	56	50	46	42	38	36	32	20	10
160	87	83	79	76	73	65	59	53	49	45	41	39	35	23	13
180	90	86	82	79	76	68	62	56	52	48	44	42	38	26	16
200	92	88	84	81	78	70	64	58	54	50	46	44	40	28	18

Steam used for direct heating in a blower or injector often gets superheated by expansion in the blower and losses can occur this way. If 30 psi.g. dry saturated steam is blown into a vat or tank containing 4 ft. 6 in. of liquor there will be a pressure reduction in the blower of 28 psi and the steam will be superheated by 38° F. It is unlikely that 4 ft. 6 in. of hot liquor can remove 38° F. of superheat and condense all the steam during its short passage through the liquor. Some steam will break the surface and be lost.

Table VI shows the superheating effect of reducing the pressure on dry saturated steam by passage through an orifice or valve.

49. DESUPERHEATING. As pointed out above and explained more fully in Chapter 5, superheated steam is often considered to be bad steam to use for heating. It may therefore be necessary to "Desuperheat" it by passing it through a desuperheater which adds a spray of distilled water to the steam. The superheat gives itself up in evaporating some of the sprayed water. In the case considered at the beginning of Section 48, the amount of superheating energy was 17 Btu. This is to be removed by evaporating water in the desuperheater. If we assume that the desuperheating water is pumped into the steam at 200° F., it must first be heated to 298° F. the boiling temperature at 50 psi.g. before evaporation can begin.

Sensible heat in water at 200° F.	168	Btu
Sensible heat in water at 298° F.	267	Btu
Difference	99	Btu
Latent heat of steam at 50 psi.g.	912	Btu

Therefore every pound of steam desuperheated will evaporate

$$\frac{17}{99 + 912} = \cdot 0168 \text{ lb. of water.}$$

This increases the weight of saturated steam leaving the desuperheater by 1·68%.

50. SUPERHEAT AND STEAM DISTRIBUTION. In certain circumstances the superheat given to steam by reducing its pressure may be very useful. Steam is sometimes very wet. If this wet steam goes into the heating surface of a piece of plant the extra condensate is just a nuisance. It has little heating value, but it coats the heating surface with an additional water film and the extra water has to be handled by the trap, and the amount of flash steam is increased.

If such wet steam can be expanded as soon as possible on its journey to the heating process, the superheat due to expansion will help to dry it. This increases the amount of steam reaching the process plant and, as the steam is at a lower pressure, it will have a higher latent heat. There may therefore be a two fold gain.

* * *

CHAPTER 2

THE POWER PROPERTIES OF STEAM

Power is more certainly retained by wary measures than by daring counsels.

TACITUS. *Annals.* 115

Chapter 1 described the effect of adding or subtracting heat energy to or from water or steam. It dealt only with steam as a heating tool. It made no mention of the mechanical power that steam can yield.

This Chapter is concerned only with steam as a power-producing tool. The standard text books deal with the subject in an alarmingly mathematical way. Apart from a little simple arithmetic, no mathematics appear in this chapter. The result is that the explanation is long, but it is hoped that it is readable, whereas a shorter mathematical description might remain unread.

51. WORK AND ENERGY. Work is done when a force operates through a distance. Work may perhaps be better described if we say that it consists of overcoming a resistance over a distance.

Energy is the capacity for doing work. Energy can be looked upon as existing in four forms :—

Potential Energy.
Mechanical Energy.
Thermal (or Heat) Energy.
Electrical Energy.

Potential energy is energy of position either of bulk as in a high level lake, or of atoms or electrons as, say, chemical energy in coal. Mechanical energy is manifested by the movement of material in bulk. Heat energy, as explained in Chapter 1, is molecular energy. Electrical energy is the energy of movement of electrons.

52. ENERGY EQUIVALENTS. Potential energy in a lake can be converted into mechanical energy. Potential energy in coal can be converted into heat energy. Heat energy can be transformed, by means of an engine, into mechanical energy which can be used for doing work. It is impossible to convert potential energy into useful mechanical energy without some form of engine. If we release the energy in coal we only get heat energy unless we interpose an engine. If we release the potential energy in a high level lake we cannot recover the mechanical energy without an engine, be it only a mill wheel. If a lake dissipates its potential energy in a waterfall, the energy is converted into heat energy and the water at the bottom of the fall will be warmer than at the high level. Mechanical energy can be converted into electrical energy by means of a generator or dynamo. Electrical energy can be converted back into mechanical energy by means of a motor, or direct back into heat energy by a radiator.

There must obviously be definite relations between the different forms of energy. These relations were first measured by a Salford brewer, J. P. Joule. On his honeymoon in Switzerland he tried to find a waterfall high enough to

give him an accurate measurement. Actually his results were obtained in another way in the laboratory, and his figures, obtained 100 years ago, have only been slightly modified. The accepted figures are given in Table VII.

TABLE VII. ENERGY EQUIVALENTS

	MECHANICAL	THERMAL	
	FOOT-POUNDS	BTU	C.H.U.
Mechanical equivalent of heat	778	1·0	·56
	1,400	1·8	1·0
Foot-ton	2,240	2·88	1·6
Foot-pound	1	·00129	·00072

53. **POWER.** Power is the rate of doing work or the rate of flow or use of energy. The normal electrical units, watts or kilowatts, are actual power units ; they express the flow of electrical energy. The equivalent electrical energy units are the watt-hour or the kilowatt-hour. Mechanical or thermal energy units have no reference to time so that mechanical or thermal power units must be energy units per unit of time ; i.e. foot-pounds per second, per minute, or per hour ; or Btu per second, per minute, or per hour.

TABLE VIII. POWER EQUIVALENTS

	MECHANICAL			THERMAL		ELECTRICAL
	FT. LB./SEC.	FT. LB./MIN.	FT. LB./HR.	BTU/HR.	C.H.U./HR.	WATTS
1 horse-power	550	33,000	1,980,000	2,545	1,414	746
1 kilowatt ..	738	44,250	2,655,000	3,415	1,897	1,000

54. **MEASUREMENT OF STEAM POWER.** When a steam engine does work, that work takes the form of a force (in pounds) operating through a distance (in feet) and the work done is measured in so many foot-pounds per minute or per hour. In a reciprocating engine this work is clearly the force in pounds acting on the piston over the distance of the stroke in feet times the number of strokes in a given time. But a turbine is more difficult. What is the force acting on each blade ? Over what distance does it act ? It would be useful to find a measuring rod that could be applied equally well to reciprocators or turbines. Of course the actual work done can only be found in either type of machine by measuring the output by means of a brake or by reading the instruments attached to an electric generator. But for considering the theory of how best to turn heat energy into work we want some good common lay-out.

A pictorial presentation of things is greatly preferable to windy explanation or a string of algebra. In Section 52 we saw that Joule found for us the rate of exchange between the energy units, so that our picture need not necessarily portray foot-pounds ; it can just as well be in Btu. In fact as the steam engine or turbine is a heat engine—that is a machine for converting heat energy into mechanical energy—it would be appropriate if the picture were to show heat units—Btu—as its *area.*

Before discussing how this picture is to be constructed a few points need clearing up.

55. EFFICIENCIES. The efficiency of an engine is given by expressing the heat energy converted into work as a percentage of the total heat energy put into the engine. This is called the THERMAL EFFICIENCY.

The efficiency of a steam cycle is the amount of heat energy that a perfect engine could convert into work expressed as a percentage of the heat energy put into the steam. Rankine was the first man to explain this idea, hence this is called the RANKINE EFFICIENCY, BASIC EFFICIENCY, IDEAL EFFICIENCY or CYCLE EFFICIENCY.

The amount of work actually done by an engine expressed as a percentage of the theoretical expected work from a perfect engine is called the EFFICIENCY RATIO.

We want to find the conditions which will give us a cycle which has the highest possible Rankine or cycle efficiency and which is such that we can construct an engine working on that cycle with the highest possible efficiency ratio. We shall thus get the highest possible thermal efficiency.

56. TEMPERATURE AS A BASIS. A steam engine or turbine is a heat engine which turns heat into work. The measure of the potential of heat is temperature. The greatest amount of work can be got out of an ideal engine when the input temperature is as high as possible and the exhaust is as cool as possible. So we should make a picture which shows energy, heat or work as its area and which is based on temperature.

57. ABSOLUTE TEMPERATURE. If temperature is to be the basis of the diagram, the temperature scale must be the correct scale. Fahrenheit took the temperature of a mixture of snow and salt as his zero. Celsius took the freezing point of water as the zero for the Centigrade scale. We all know that in Russia or Canada, or at 30,000 ft. in an aeroplane anywhere, temperatures much below these zeros are met.

It has been found by numerous experiments that gases expand and contract when they are under constant pressure by an exact and unchanging amount for any particular temperature change.

This amount is $\frac{1}{492}$ for 1° F. change at 32° F.

Suppose we have 492 cu. ft. of any perfect gas at 32° F.

If the gas is cooled under constant pressure by 1° F. to 31° F. the volume will contract to 491 cu. ft.

If the temperature of the gas is reduced to 0° F., keeping the pressure constant, the volume will contract to 460 cu. ft.

This holds good for all so-called perfect gases over the range that experiment permits. The only possible conclusion is that at −460° F. any perfect gas at any pressure will have no volume. Actually, of course, at very low temperatures gases behave in anything but a perfect manner ; they liquefy or solidify —which is perhaps just as well for the theory which might otherwise get into difficulties. There are other considerations which point to −460° F. as being the lowest possible temperature or ABSOLUTE ZERO. It has been explained in Chapter 1 that gases consist of molecules in motion. The hotter the gas, the more violent the motion. It can be proved theoretically that this motion would cease at −460° F. So our diagram will be based on a temperature scale starting at −460° F. and will be an "Absolute Temperature" scale. The area on the diagram will be energy in Btu.

58. CONSTRUCTION OF THE DIAGRAM. We will now construct the diagram. The description is long, but is quite elementary and easy to follow.

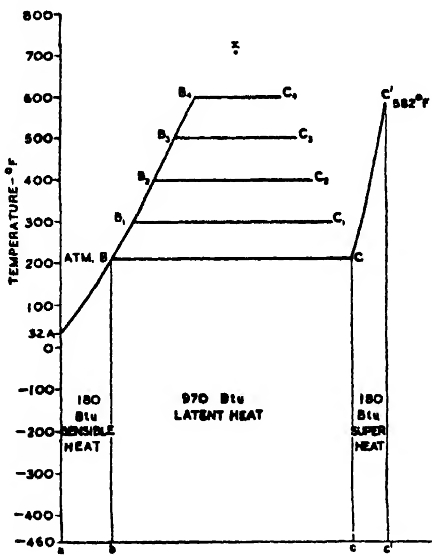


FIG. 4. CONSTRUCTION OF STEAM POWER DIAGRAM

Set up a vertical scale, Fig. 4, from − 460° F. to say + 1000° F. A convenient skeleton diagram will be obtained if we use a scale of 250° F. to the inch and 200 Btu to the square inch. (The diagrams here are much smaller than this.)

At point A the water is about to freeze and is considered to have no heat content. (This of course is not true—see Section 794. Freezing water or ice

or anything above $-460°$ F. has a heat content, but ice is no use in a heat engine, so water at freezing point is taken as the starting point and is considered to have no heat.)

Now the diagram is to show the properties of 1 lb. of water and/or steam. If we add 180 Btu to 1 lb. of water at 32° F. under atmospheric pressure we shall raise the water to boiling temperature of 212° F., point B in Fig. 4.

We know the height of point B, but we want to know the distance a b such that the area A B b a will be equal to 180 Btu.

A rectangle of area equal to A B b a will have a height half way between A and B, or 122° F.

So the height of the equivalent rectangle will be $122 + 460 = 582°$ F. Abs.

As each inch is equal to 250° F., the height of the rectangle will be 2·328 in.

As each square inch of area is to represent 200 Btu, 180 Btu will be equivalent to ·9 sq. in.

The width a b will be $\frac{\cdot 9}{2\cdot 328} = \cdot 387$ in.

We can now fix point B at 212° F. and ·387 in. to the right of A a.

To vaporise 1 lb. of water at 212° F. under atmospheric pressure requires the addition of 971 Btu of latent heat. This takes place at constant temperature, so it is represented by the rectangle B C c b.

The height is $212 + 460 = 672°$ F. Abs. or 2·688 in.

The area of 971 Btu will be 4·85 sq. in.

Therefore the width b c will be $\frac{4\cdot 85}{2\cdot 688} = 1\cdot 804$ in.

This gives us point C.

Line A B C represents the state of 1 lb. of water during the absorption of 1,151 Btu of heat energy which changes it from water at freezing point into saturated steam at atmospheric pressure.

If we repeat this performance at higher pressures, and consequently higher temperatures, we get lines A B_1 C_1, A B_2 C_2, etc. Each of these lines shows the effect of adding heat energy to freezing water and completely vaporising it at various pressures and their corresponding temperatures. All the necessary figures are in the saturated steam tables.

Area A B b a represented the addition of a relatively large amount of heat and we took A B as being a straight line. Actually line A B B_1 B_2, etc., must be curved because the temperature increases caused by adding equal amounts of heat become gradually smaller as the temperature becomes higher. The actual curve can be found by taking a large number of small heat additions.

At 705·6° F. at a pressure of 3,208 psi.a. latent heat disappears, as explained in Section 42. This is the " Critical Point " and is shown as point X.

In this Chapter almost all the pressures used in the discussion are absolute, not gauge pressures. The reason for this will be found later.

59. BOUNDARIES. We can now draw a curve through A B B_1 B_2 B_3 B_4 X C_4 C_3 C_2 C_1 C. This is called the "Boundary Curve" and is shown in Fig. 5. Every point underneath the curve represents a water-steam mixture at a particular temperature and pressure. This part of the diagram is called the Wet Region. The left-hand part of the boundary curve A B B_1 B_2 B_3 B_4 X in

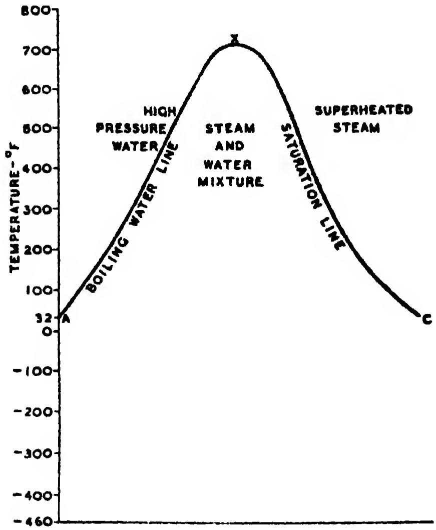


FIG. 5. BOUNDARY CURVE

Fig. 4, shows the boiling points of water corresponding to various pressures, and is called the Water Line. The right-hand part of the curve, Fig 5, represents the states of dry saturated steam at various pressures and is called the Saturation Line. All the points to the right of the saturation line represent the states of steam containing more heat energy than in the saturated state. Such steam must therefore be superheated. This part of the diagram is called the Superheat Region. The area to the left of the water line represents water at high temperature yet containing insufficient heat to be at boiling point. Water in this area must be at a pressure above the boiling pressure corresponding to the temperature. This area is called the Water Region.

Fig. 4 shows that the heat in steam at high temperature occupies a taller narrower area than the heat in steam at lower temperatures. The energy is more concentrated, it is not so spread. Every heat addition at higher temperature is a taller thinner slice.

60. SUPERHEAT. Point B in Fig. 4 shows the state point of water after the addition of 180 Btu of sensible or liquid heat. Point C′ represents the state point of steam after the addition of 180 Btu of superheat to dry saturated steam at C.

The steam table tells us that steam containing 180 + 970 + 180 = 1,330 Btu per pound at atmospheric pressure will have a temperature of 590° F., so that fixes the height of point C′.

The rectangle of area equal to C C′ c′ c will have a height of

$$\frac{(590 + 460) + (212 + 460)}{2} = 861° \text{ F. or } 3{\cdot}45 \text{ in.}$$

The additional area is 180 Btu or ·9 sq. in. so that the width or spread c c′ will be

$$\frac{{\cdot}9}{3{\cdot}45} = {\cdot}261 \text{ in.}$$

So point C′ is on the 590° F. line and its distance from A a is

$${\cdot}387 + 1{\cdot}804 + {\cdot}261 = 2{\cdot}452 \text{ in.}$$

This of course shows C C′ as a straight line. It is actually a curve tending towards the vertical for the same reason as the water line curves, and its shape can be found by taking sufficiently small heat additions.

61. PRESSURE LINES. The foregoing sections, 58–60, have shown how a pressure line, for example atmospheric pressure, can be drawn in. By taking the values for various pressures from the steam tables we can draw in as many pressure lines as are convenient. Some of these are shown in Fig. 6, including some below atmospheric pressure.

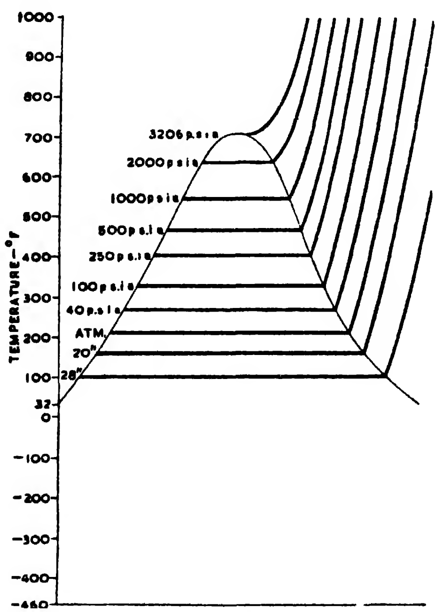


FIG. 6. PRESSURE LINES

62. HEAT ADDITION AT HIGH AND LOW PRESSURE. It is very interesting to compare the raising of steam at high temperature and pressure with the addition of heat at low temperature and pressure. Let us raise steam at 1,325 psi.a. and superheat it. Let us do this by adding instalments of 59 Btu at a time. See Fig. 7. At 1,325 psi.a. the sensible heat is 590 Btu and

the latent heat is also 590 Btu. Let us then add $3 \times 59 = 177$ Btu of superheat. The total heat added will have been $590 + 590 + 177 = 1{,}357$ Btu. As the heat is being added in 59 Btu instalments there will be 23 instalments which appear on the diagram as vertical strips. It will be seen that the higher the temperature at which the heat addition is made, the narrower is the strip—the less the energy is spread.

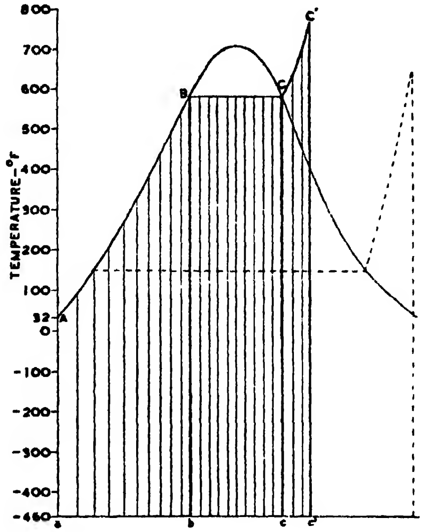


FIG. 7. HEAT ADDITION AT HIGH TEMPERATURE AND PRESSURE

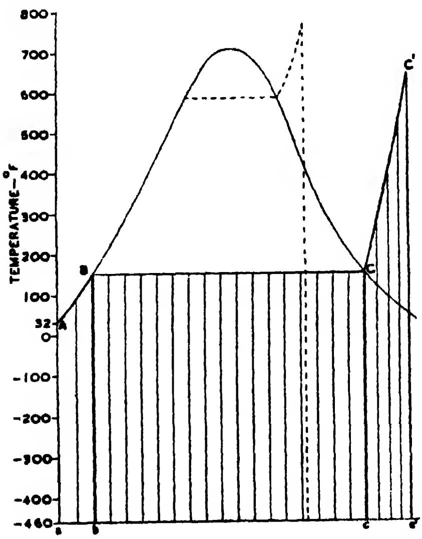


FIG. 8. HEAT ADDITION AT LOW TEMPERATURE AND PRESSURE

Now let us raise steam at 22½ in. vacuum, Fig. 8. The sensible heat required is 118 Btu, the latent heat is 1,008 Btu and the total heat at saturation is 1,126 Btu. If we add the same amount of heat as in Fig. 7 we must add 1,357 − 1,126 = 231 Btu of superheat. The heat is added, as in Fig. 7, in instalments of 59 Btu. Comparison of Figs. 7 and 8 shows how much wider are the heat addition strips in Fig. 8 and how the whole diagram is spread to the right. In each of Figs. 7 and 8 the pressure line for the other diagram is shown dotted for comparison.

In Fig. 7 we produced superheated steam at 1,325 psi—fine, powerful stuff. In Fig. 8 the steam contains exactly the same amount of heat energy, but the heat is spread over the huge, tenuous volume of 22·3 in. vacuum vapour. Although this vapour is superheated to nearly 650° F. it has very little virtue. It is very clear that when we add heat to steam for power purposes we should add the heat in as concentrated a form as possible—we should keep the spread to a minimum.

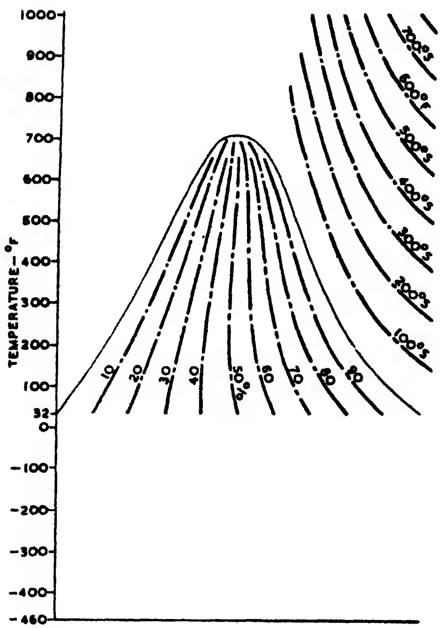


FIG. 9. QUALITY LINES—PER CENT. DRYNESS AND CONSTANT SUPERHEAT

63. HEAT SUBTRACTION. When heat is added to steam or water it is added to the diagram in the form of vertical slices clapped on to the right-hand side of the diagram. When steam or water is cooled, the heat is taken away in vertical slices from the right-hand side of the diagram. Superheat must be removed before condensation can take place, as explained in Section 22, and Figs. 4, 7 and 8 show that this entails taking slices off the right-hand side of the diagram. When the superheat has been removed further cooling

takes vertical strips off the right-hand side of the picture and is the exact opposite of the heat addition shown in Figs. 7 and 8. Such strips represent the removal of latent heat and cause condensation, the amount of which can be measured as wetness along the horizontal evaporation pressure line.

64. WETNESS LINES. In Fig. 7 we added ten strips of heat between B and C, each of 59 Btu, to vaporise completely 1 lb. of water at boiling point at 1,325 psi. Each strip must have evaporated 10 per cent. of the water. So that after adding nine strips we must have had 90 per cent. steam and 10 per cent. water. By dividing each horizontal pressure line (that is that part of the line where evaporation only is taking place) into ten parts and joining the corresponding points on each pressure line we can get lines of constant wetness or constant dryness, whichever we like to call them. These quality lines are shown in Fig. 9.

65. SUPERHEAT LINES. By measuring off equal temperature rises along the pressure lines in the superheat region, these points can be joined by curves which are lines of constant superheat. Some are shown in Fig. 9.

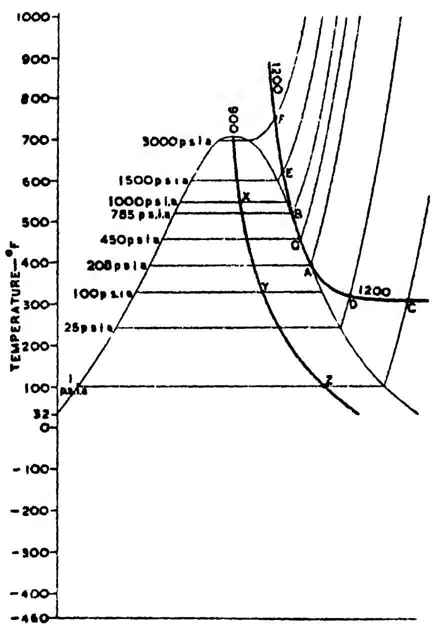


FIG. 10. PLOTTING TOTAL HEAT LINES

66. TOTAL HEAT LINES. Every point on the diagram shows the state of steam containing a particular amount of heat energy at a particular pressure. This heat energy is represented by the area below the constant pressure line on which the point lies. There must be many points on the diagram which have the same total heat value; for example, points C′ on Figs. 7 and 8 each represent superheated steam containing 1,357 Btu, but in very different state otherwise.

Suppose we wish to draw in the line passing through all the state points where the steam contains 1,200 Btu/lb. The saturated steam table tells us that dry saturated steam contains 1,200 Btu at 208 psi.a. and at 785 psi.a. These are shown on Fig. 10 as points A and B. The superheat table tells us that steam contains 1,200 Btu at the following temperatures and pressures in the superheat region :—

1 psi.a., 309° F., point C.
25 psi.a., 319° F., point D.
1,500 psi.a., 617° F., point E.
3,000 psi.a., 753° F., point F.

Lines can be drawn for total heat in the wet region thus : Assume we wish to draw the 900 Btu line. Inspection of the steam table shows that in no condition has dry saturated steam as little as 900 Btu. All the points on the 900 line must therefore indicate a mixture of water and steam.

The sensible heat of water boiling under 1 psi.a. (28 in. vac.) is 69·7 Btu.

The latent heat of steam at this pressure is 1036·3.

Therefore the latent heat in 1 lb. of steam-water mixture containing 900 Btu is $900 - 69\cdot7 = 830\cdot3$.

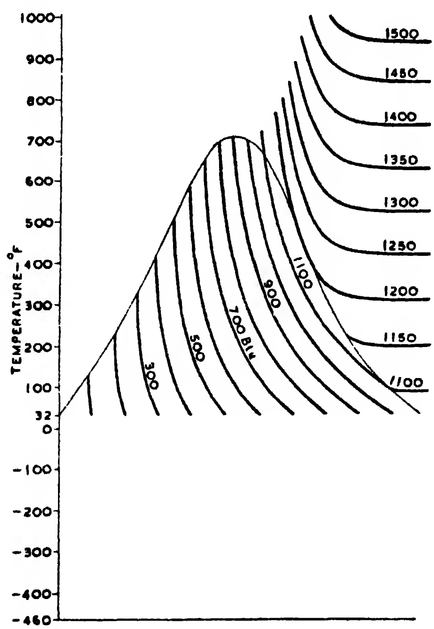


FIG. 11. TOTAL HEAT LINES

The amount of steam in the mixture must be $\frac{830\cdot3}{1036\cdot3}$ 0·8 lb. or 80 per cent. steam.

This is shown as point Z in Fig. 10.

At 100 psi.a. the sensible heat is 298·4 and the latent heat is 889·6, so that the 900 Btu total heat point will consist of

$$\frac{900 - 298 \cdot 4}{889 \cdot 6} = \cdot 676 \text{ lb. or } 67 \cdot 6 \text{ per cent. steam.}$$

This is point Y in Fig. 10.

Similarly point X is found to be 55 per cent. steam at 1,000 psi.a.

The 1,200 Btu line can be completed between points A and B in a similar way. Point G shows the 1,200 point at 450 psi.a. corresponding to 99·4 per cent. dryness. Any desired total heat lines can thus be drawn, and some of these are shown in Fig. 11. It will be seen from line F E B G A D C in Fig. 10 that saturated steam has a maximum total heat (nearly 1,206) at 450 psi.a.

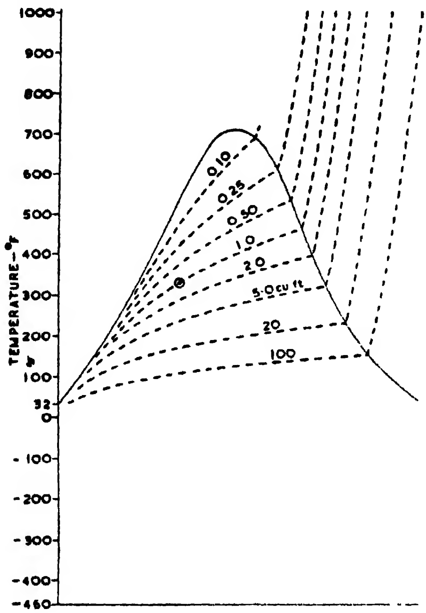


FIG. 12. VOLUME LINES

67. VOLUME LINES. The steam tables show the volumes occupied by steam and water under various pressures and temperatures. It is therefore possible to draw lines of constant volume on the diagram ; for example, lines showing all the states where steam and/or water occupy 1·0 cu. ft. or 10 cu. ft. or 100 cu. ft. Suppose we wish to find the point where the volume is 1 cu. ft. at 100 psi.a. The steam table tells us that 1 lb. of water at 100 psi.a. occupies ·0177 cu. ft. and that 1 lb. of saturated steam occupies 4·432 cu. ft. Clearly the desired state will be one where there is about 1/5 steam and 4/5 water.

Let x = dryness fraction.

Then $4 \cdot 432\, x + (1 - x) \cdot 0177 = 1$.

Whence $x = \cdot 221$ or 22·1 per cent. dry.

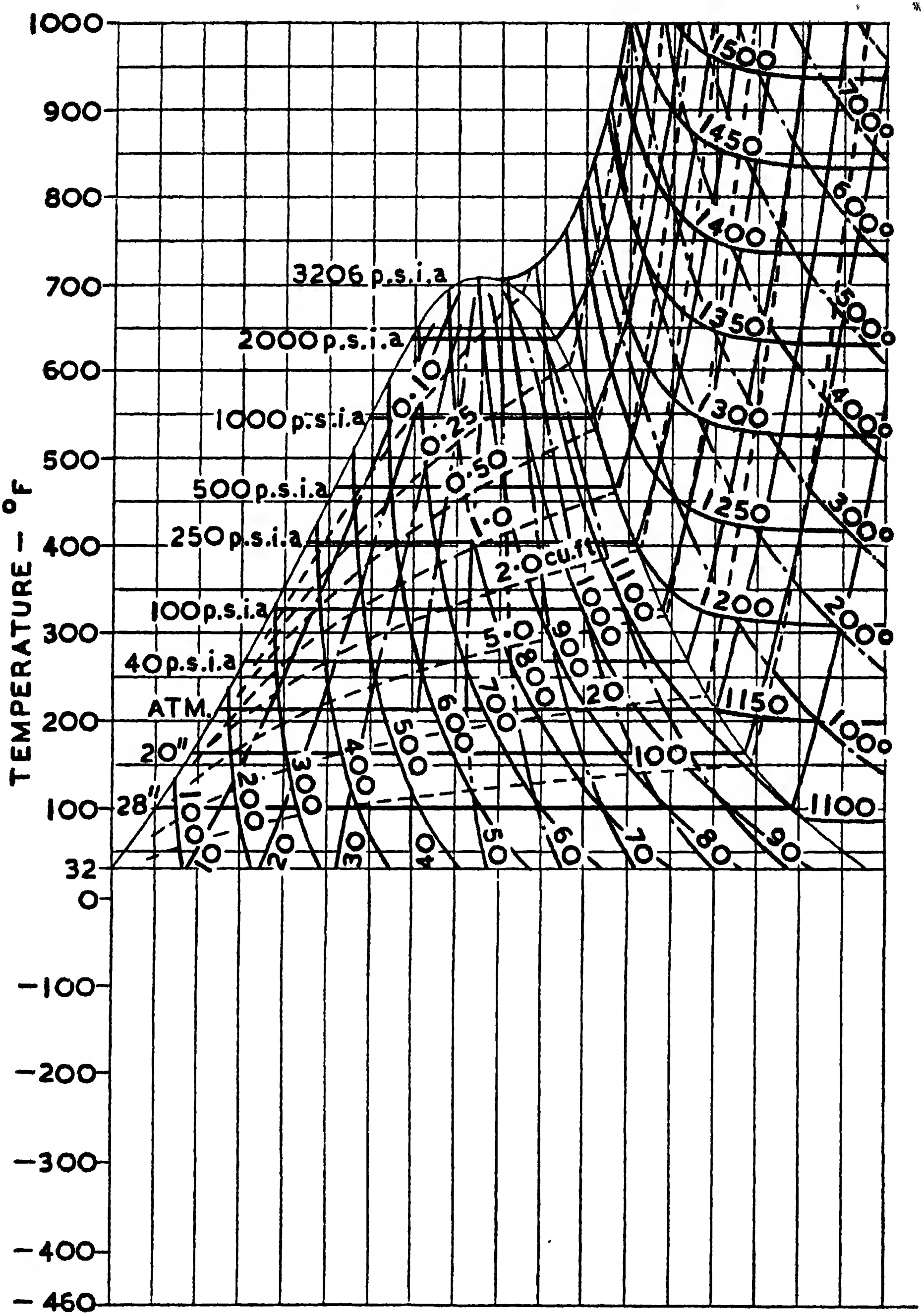


FIG. 13. COMPLETE STEAM POWER DIAGRAM

This is shown as point o in Fig. 12 which shows a number of the constant volume lines. The points for the volume lines in the superheat region can be found in the superheat table.

68. THE COMPLETE DIAGRAM. We can now superimpose all the lines shown in Figs. 5, 6, 9, 11 and 12 on one picture which will thus show all the properties of steam and water and is shown in Fig. 13. Had we approached this diagram from the start we might well have been dismayed by its complexity. All the properties of steam at any state can be read off the diagram, but, what is much more important, we can see exactly what happens when we make steam change its state in any particular way—if we change its pressure or volume by compressing it or letting it expand ; if we change its temperature or heat content by cooling it or heating it, or by doing work on it or taking work out of it.

A good complete diagram is contained in " The Thermodynamic Properties of Steam " by Keenan and Keyes—see Section 805.

69. CONSTANT PRESSURE CYCLE. Let us now carry out certain imaginary operations and see what happens. Let us assume that we possess the curious piece of apparatus shown in Fig. 14. This consists of a tall vertical cylinder containing an engineer's dream—a weightless, frictionless, leak-proof piston. In the bottom of the cylinder is 1 lb. of water on which the piston

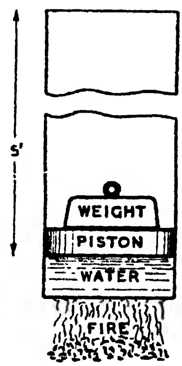


FIG. 14. IMAGINARY STEAM CYLINDER

is resting. The area of the piston is 1 sq. ft. or 144 sq. in. The bore of the cylinder will therefore be just over $13\frac{1}{2}$ in.

The pressure of the atmosphere, 14·696 psi is acting on the piston top producing a load on the water of 2,116 lb. Let us place a weight of 12,284 lb. on the piston. The total load resting on the water is therefore 14,400 lb. or 100 psi. Let us assume that the water is at a temperature of 212° F.

Let us put a source of heat—a portable fire—under the cylinder and heat the water. The boiling point of water at 100 psi.a. is 328° F. and such water contains 298 Btu of sensible heat. The water at 212° F. already contained

180 Btu so that we had to add 298 − 180 = 118 Btu to raise the water to its 100 psi boiling point. This operation is shown in Fig. 15 as a movement of the state point from A to B by the addition of heat area A B b a.

Continued addition of heat evaporates the water at 100 psi along line B C. As soon as evaporation starts the piston will begin to rise in the cylinder. Complete evaporation requires the addition of 889 Btu and is shown as area B C c b. The steam table tells us that the volume of 1 lb. of saturated steam at 100 psi.a. is 4·432 cu. ft. So that the volume has increased from ·0167 cu. ft. (the volume of 1 lb. of water at 100 psi and 212° F.) to 4·432 cu. ft. As the cylinder has a cross section of 1 sq. ft. the piston will have risen 4·432 − ·0167 = 4·4153 ft. or 4 ft. 5 in. It has overcome a resistance of 14,400 lb. over the distance of 4 ft. 5 in. and therefore the heat addition which has changed the water into steam at 100 psi has done work equal to 14,400 × 4·4153 = 63,580 ft. lb.

Now remove the fire and cool the cylinder. Condensation will commence and the piston will fall. The state point on Fig. 15 will move from C towards B. When 889 Btu have been removed, the state point will have moved back to B, and the piston will be resting on 1 lb. of water at 328° F. Further cooling will remove the sensible heat until, when 118 Btu have been removed, the state point will be back at A and the cycle will have returned to its starting point. All the work done in raising the weight 4 ft. 5 in. has been dissipated by allowing heat rejection to take place at the same pressure as the heat addition. Clearly such a constant pressure cycle is useless for the production of power.

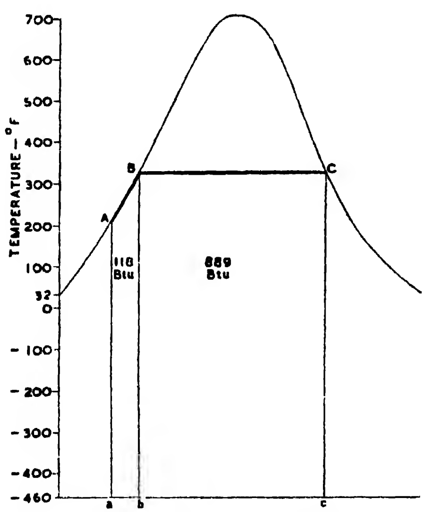


FIG. 15. CONSTANT PRESSURE CYCLE

70. VARYING PRESSURE CYCLE. Let us repeat the experiment making two small modifications to the apparatus. Fix a stop in the cylinder to limit the piston travel to 4 ft. 5 in., and provide a chain with a hook on the end of it above the cylinder in such a way that the weight can be attached to the hook. Now add heat as before to the water in the cylinder at 212° F. until it is all

vaporised at 100 psi.a. The piston will rise 4 ft. 5 in. overcoming a resistance of 14,400 lb. The heat addition is area A B C c a, in Fig. 16 and is 118 Btu of sensible heat + 889 Btu of latent heat = 1,007 Btu. Now let us hook the weight to the chain, pull the chain taut, and secure it. The chain does not yet actually support the weight as this is held in position by the 100 psi pressure of the steam inside the cylinder. Now remove the fire and cool the cylinder. Condensation will start, the pressure will drop and an increasing proportion of the weight load will be taken by the chain. This process will continue until the amount of heat removed brings the pressure in the cylinder down to atmospheric pressure, and the chain will be carrying the whole of the load of the weight. The piston is, however, still in contact with the weight, because the pressure below the piston is the same as that above it, namely atmospheric pressure and we have assumed that the piston has no weight of its own. Under these conditions all the heat so far rejected has been lost at constant volume. The state point must, therefore, have been travelling back along the constant volume 4·432 cu. ft. line C D towards D in Fig. 16. Continued cooling of the cylinder will tend to drop the pressure below atmospheric, but as the atmosphere is acting on the top of the piston, it pushes it down so as to maintain the pressure inside the cylinder equal to that outside. Consequently this part of the process must be taking place at constant pressure, along line D A. The piston will fall until condensation is complete at atmospheric pressure when the piston will be resting on 1 lb. of water at 212° F., and we are back at our starting point. But—the weight is hanging on the chain 4 ft. 5 in. higher than it was at the start of the cycle.

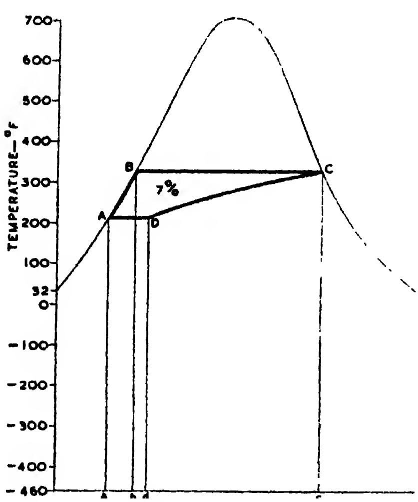


FIG. 16. NON-EXPANSIVE WORKING

Now the heat input was area A B C c a = 1,007 Btu.

The heat rejected was area C D A a c, which we can measure either with a planimeter or by splitting into rectangles and triangles or by "counting squares" if our diagram has been drawn on squared paper.

We find, by the latter method, that it is equal to 937 Btu.

We have therefore used 1,007 − 937 = 70 Btu to raise a weight of 12,284 lb. by 4 ft. 5 in.

This is shown in Fig. 16 as area A B C D.

The work done, and stored in the hanging weight, is 12,284 × 4 ft. 5 in. = 54,238 ft. lb.

Now we see from Table VII in Section 52 that 778 ft. lb. = 1 Btu so that 70 Btu will equal 54,460 ft. lb. A good agreement, which showed that we measured our area well.

The total heat input was 1,007 Btu.

We did useful work equivalent to 70 Btu.

So this cycle has a Rankine or cycle efficiency of 7 per cent.

71. EXHAUST TO ATMOSPHERE. In the cycle just described in Fig. 16 the heat was rejected by cooling the cylinder and compelling the state point to move along the path C D A. Had we simply blown the exhaust to atmosphere we should still have rejected the same amount of heat, area A D C c a, but instead of being left with water at 212° F., point A, we should have blown away our hot water containing 180 Btu (the strip to the left of A a) as well.

72. HEAT REJECTION PATH. Heat can always be regarded as being rejected from an engine along the constant volume line down to the exhaust or back pressure, and then along the back pressure line to the water line, and then down the water line if the water is rejected as well.

73. REVERSIBLE CYCLES. Let us return to our cylinder which we left at the end of Section 70, with the weight hanging 4 ft. 5 in. above the piston which is resting on 1 lb. of water at 212° F.—point A in Fig. 16. Without touching anything let us apply heat to the cylinder. As the water is at atmospheric pressure and at 212° F., evaporation will start at once and the piston will rise. The first part of the evaporation will be done at constant (atmospheric) pressure because there is no weight on the piston except the pressure of the atmosphere. The state point will move along A D, Fig. 16. At point D the piston will have risen 4 ft. 5 in. and will be in contact with the weight. Continued evaporation will take place at constant volume and the pressure will rise so that an increasing amount of the load of the weight will be borne by the piston. When all the water has been evaporated the pressure will have become 100 psi.a. ; the state point will have moved up the constant volume line from D to C, and the whole of the load of the weight will now be carried by the piston. Now unhook the chain from the weight. The weight will not drop, as there is sufficient pressure below the piston to sustain it. Remove the fire and cool the cylinder. Heat rejection must take place at the constant pressure of 100 psi.a. under the pressure of the weight and of the atmosphere, and the state point will move along the constant pressure line from C towards B. When all the 889 Btu of latent heat have been removed (area C B b c) the piston will be resting on 1 lb. of water at 328° F., point B. Continued cooling will remove sensible heat from the

water, 118 Btu, and the state point will have returned to the original position at A of water at 212° F.

We added heat area A D C c a which we know is 937 Btu. We allowed the potential energy in the weight to act on the steam and become mechanical energy and so do work of compression equivalent to 70 Btu. We rejected, by cooling, area A B C c a = 1,007 Btu.

Input Heat 937 + Work 70 = Rejected 1007.

By adding 70 Btu of mechanical energy of compression to 937 Btu of heat energy we produced (for rejection) 1,007 Btu. In Section 70 we showed that by expansion from water we could capture 70 Btu for work from the addition of 1,007 Btu, but that we had to reject 937 Btu.

Input Sensible Heat 118 + Latent Heat 889 = Work 70 + Rejected 937.

We see therefore that the cycle is reversible. Now it is only possible to get the maximum amount of mechanical work out of heat energy if a reversible process is used.

The more nearly the process is to being completely reversible, the more power can a given amount of energy yield. Contrariwise, the more irreversible the process the more power is irretrievably wasted.

The difference between a reversible process and an irreversible process is sometimes obscure. A steam engine and boiler is a combination of reversible and irreversible processes. The engine can work on a reversible cycle, but the boiler is irreversible. By pumping steam into the boiler by reversing the engine we cannot by any conceivable means produce a fire ! A rocket works on an irreversible plan. A mill wheel is reversible, but a waterfall is irreversible. Nothing can possibly make water climb up a waterfall. If, however, the water falls inside the buckets of a water wheel, a drum on the wheel spindle can be made to wind up a weight. If the weight is allowed to fall, it will turn the wheel the wrong way and the buckets will scoop up water and lift it to the level of the mill pond.

74. IRREVERSIBLE PROCESS. In a steam boiler we generate steam under constant pressure although the boiler has a constant volume, because the steam as it is generated is piped off for use. Let us take a pound of 100 psi.a. steam which has been raised in a boiler from water at 212° F., along line A B C in Fig. 17. It is at state C occupying 4·432 cu. ft. and contains 1,188 Btu. Blow this steam into a completely empty vessel which has a volume of 30 cu. ft. The pressure will drop, the volume will increase, but there has been no change of energy. The steam has only done work on itself by expanding itself. If the vessel is assumed to be perfectly lagged, all the 1,188 Btu are still there, so the state point must have moved along the constant total heat line from C to C′ on the 30-cu. ft. volume line. The diagram shows that this point is at atmospheric pressure. Inspection of the steam table will confirm that the only state of steam containing 1,188 Btu and occupying 30 cu. ft. is at atmospheric pressure superheated to 289° F.

We put in heat equal to area A B C c a. We have done no work. We have rejected nothing. By useless expansion we have increased the spread from c to c′. Area A y C′ c′ a is equal to area A B C c a. As area A x c a is common

to both, area A B C x must equal area x y C′ c′ c. By useless expansion, we have taken the horizontal energy slice A B C x and clapped it on to the right-hand side as a vertical spread-increasing slice x y C′ c′ c.

Can we get steam at state C′ back to steam at state C ? The only possible way might be by compression. But compression will add energy, whereas steam at state C has no more energy than steam at state C′. So this process is not reversible and we have seen that no work was obtained from it. Let us say it again : " Maximum work can only be obtained from a completely reversible cycle ".

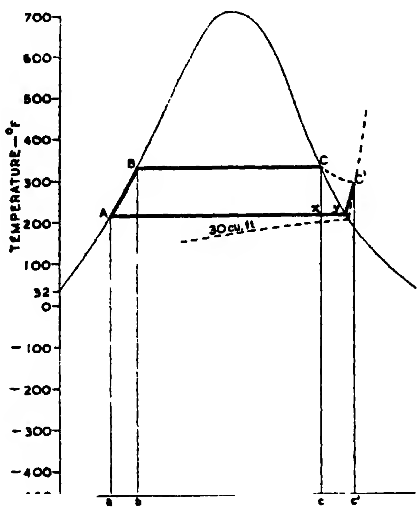


FIG. 17. WIREDRAWING

Fig. 17 should be compared with Figs. 7 and 8. The process used in Fig. 8 was the addition of heat to produce superheated steam at 22·33 in. vac. at C′, containing 1,357 Btu. Exactly the same steam would have been obtained had we blown steam at 1,325 psi.a. superheated to 770° F., Fig. 7, into an empty vessel of such a size that the pressure would have been reduced to 22·33 in. vac. The state point C′ in Fig. 7 would have moved along the 1,357 constant total heat line to C′ in Fig. 8.

75. PATH OF STATE POINT. If, after expanding our steam from C to C′ in Fig. 17, we cool the 30-cu. ft. container and reject the heat represented by area A y C′ c′ a, we shall be left with water at 212° F., point A, whence we started. But area A y C′ c′ a is equal to area A B C c a, so we have done no work. In Figs. 16 and 17 as we heat and expand, cool and reject, the state point traces out an area—A B C D in Fig. 16 ; area A B C C′ y in Fig. 17. In Fig. 16 the area A B C D was the work done, but no work has been done in Fig. 17. So here is another rule : " In a completely reversible cycle, the area traced out by the state point shows the work available ". In an irreversible or partly irreversible cycle the area traced out by the state point has no exact meaning.

76. STATE POINT PROPERTIES UNVARYING. The state points C′ of steam in Figs. 8 and 17 show the properties of steam at those points. These properties are exactly the same whether the steam be generated at high pressure and blown down into a bigger volume, or raised at low pressure and then superheated. The steam at point C′ in Fig. 17 has 180 Btu of sensible heat, 970 Btu of latent heat, and 38 Btu of superheat. We actually obtained steam at state C′, Fig. 17, by taking water with 180 Btu of sensible heat, adding a further 118 Btu of sensible heat, adding 889 Btu of latent heat and then expanding at constant total heat. Can we say therefore what are the individual heat components of steam at any state ? The recognised and logical way is to assume that the heat content of steam is made up of those heat additions that are needed to make the steam reach its particular state point by the lowest reversible path, i.e. along the constant pressure line.

77. EXPANSIVE WORKING. Let us return to our experimental cylinder. We are now going to carry out an operation that will require considerable modifications to the apparatus. The cylinder must be about 25 ft. high. The piston must carry an extension rod about 21 ft. long on it. On the top of the rod is a platform to carry the weight. Platform, rod and piston are assumed

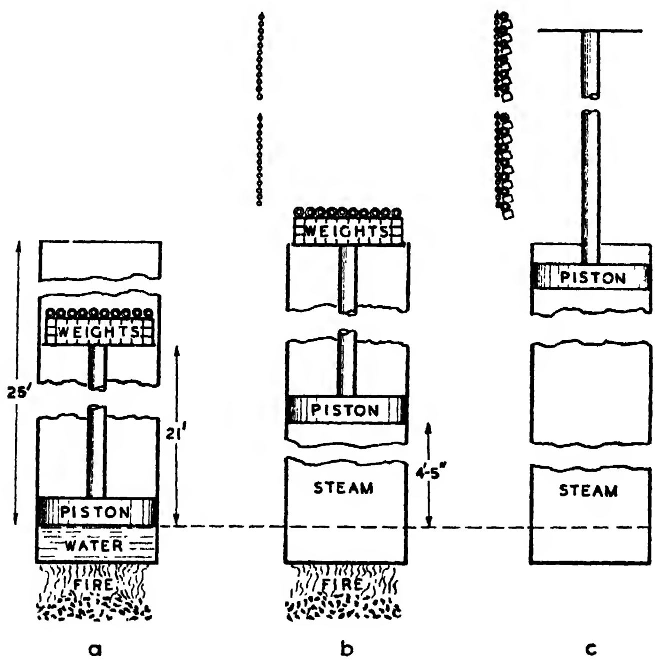


FIG. 18. MODIFIED IMAGINARY CYLINDER

to have no weight. Fig. 18a shows the new arrangement. It is assumed in all these experiments that the cylinder is perfectly lagged except at its base and that it suffers no radiation loss through the sides or through the piston.

Let us start as before with the piston resting on 1 lb. of water at 212° F. Let the weight on the piston again be 12,284 lb., but made up of a multitude of little weights instead of one big weight. Apply heat to the cylinder base. Again the water will be heated to 328° F., the boiling point at 100 psi.a., after the addition of 118 Btu. The state point moves from A to B, Fig. 19, with the addition of heat area A B b a. During evaporation along B C the piston will rise, lifting the weights. At C when the evaporation is complete, the piston will have lifted the load 4 ft. 5 in., when 889 Btu of latent heat, area B C c b, Fig. 19, have been added. This is the position shown in Fig. 18b.

Now remove the fire and cover the base of the cylinder with perfect lagging. Remove one small weight from the platform and attach it to the chain. Remove the stop in the cylinder which limited the piston movement to 4 ft. 5 in. Owing to the removal of the small weight, the piston will rise slightly until the pressure inside the cylinder is balanced by the reduced load on the piston. Remove another small weight and hook it on the chain. The piston will rise again. Carry out this small-weight-removing process until all the weights have been hung on the chain. There will now be only the pressure of the atmosphere pressing on the top of the piston, and, as the piston is assumed to be of no weight, the pressure inside the cylinder will be atmospheric pressure also, Fig. 18c.

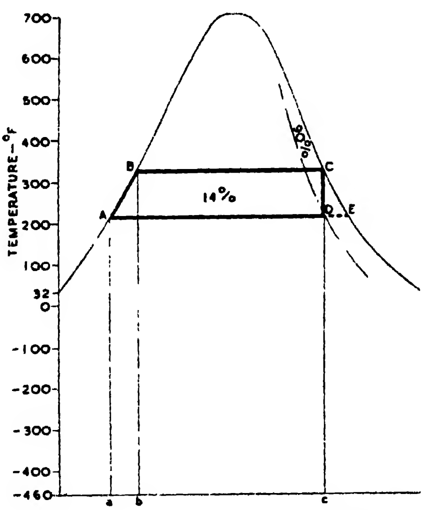


FIG. 19. EXPANSIVE WORKING

Clearly, as the long string of weights proclaims, we have done much more work than the raising of the single weight 4 ft. 5 in. What has happened to the steam ? How has the state point moved ? We know that loss of heat by cooling moves the state point to the left. We know that increase of volume by useless expansion moves the state point to the right. We have done neither of these things, so that the state point must have moved downwards only. We know that the steam left in the cylinder has lost energy because we have a store of energy given up by the steam in our string of weights.

We know that the steam has lost pressure and temperature. We know that it has increased in volume. All these things happen when the state point moves vertically downwards. So that during work-producing expansion the state point moves in Fig. 19 from C to D on the final pressure line—in this case atmospheric pressure. A little consideration will show that any deviation from the vertical indicates imperfection in the cycle. If the line slopes down to the right there must have been useless expansion or wiredrawing; if the line slopes down to the left, condensation must have taken place. (If we were crazy we might devise a process combining wiredrawing and condensation which would make the state point move from C to D without doing work.)

We can now reject the heat remaining in the cylinder by cooling, so getting rid of heat area A D c a, when the piston will return to its original position resting on the water at 212° F.

Is the process reversible? We can add heat to the water at 212° F. Evaporation will take place at atmospheric pressure pushing the piston up until the platform is level with the topmost weight. This will require the addition of heat area A D c a. Then, by adding one weight at a time, we can drive the piston down until we have recompressed the steam from D to C at 100 psi.a. By cooling we can now reject the heat at constant pressure along line C B A. The cycle is therefore completely reversible so that the area traced out by the state point, A B C D, is the measure of the work done. Area A B C c a is 1,007 Btu. Area A D c a is easily measured and is 867 Btu. By difference, area A B C D is 140 Btu. This gives a Rankine Efficiency of 14 per cent. and represents 108,920 ft. lb. stored in the string of weights.

78. EXTERNAL WORK AND INTERNAL ENERGY. What is the essential difference between the two cycles shown in Figs. 16 and 19? In Fig. 16 the steam has merely been generated and during generation has done work. In Fig. 19 the steam has not only been generated but has been usefully expanded or "used" and the work done was of two kinds—first the work done by expanding the water into steam and second by allowing the steam to expand against a work-producing piston. This tells us something important. External work is done during evaporation at constant pressure due to the increase in volume.

In Fig. 16 the pressure was 100 psi.a., the piston area was 1 sq. ft. or 144 sq. in., the volume increase was 4·415 cu. ft., on an area of 1 sq. ft., a linear expansion of 4·415 ft.

So that the external work done during the production of steam from the water was $\dfrac{100 \times 4{\cdot}415 \times 144}{778} = 82$ Btu.

The difference between the total energy in steam at 100 psi.a and the external work done during evaporation is the "Internal Energy" of the steam. (The values of internal energy are sometimes given in steam tables.)

In Fig. 16 we did not capture all the external work because the overcoming of the atmospheric pressure accounted for some of it. The capturable or net work done during steam generation is proportional to the difference between the pressure at which the steam is raised and the back pressure.

In Fig. 16 this difference is the gauge pressure at the start and atmospheric pressure at the end.

The net work done in Fig. 16 was $\frac{(100 - 14\cdot7) \times 4\cdot415 \times 144}{778} = 70$ Btu.

In Fig. 16 we generated steam, captured its net external energy of evaporation and rejected the whole of its internal energy. In Fig. 19 we got 70 Btu of external work and then used the steam by expansion against a piston to secure another 70 Btu of work from the internal energy of the steam.

Although a boiler has a constant volume, it works as a constant pressure machine because the steam is piped off as fast as it is generated and does its external work of power production in the engine at the other end of the pipe.

A direct-acting pump operates on the non-expansive cycle shown in Fig. 16 and only uses the external work done by the steam during its production from water. A turbine, or an engine fitted with cut-off valve gear, works on the expansion cycle shown in Fig. 19 and uses some of the great store of internal energy in the steam. We must see how the cycle can be so improved as to use as much as possible of this internal energy, but there are certain points that we can usefully consider before we set out to improve the cycle.

79. EXHAUST STATE—HEAT DROP. Point D in Fig. 19 shows the state of the steam at the exhaust point. It shows that the steam is 11 per cent. wet, that it has a volume of 24 cu. ft. and that it has a heat content of 1,048 Btu. Now at point C the total heat content (read off the diagram or found in the steam table) was 1,188 Btu, so that the steam has suffered a "Heat Drop" during expansion of 1,188 – 1,048 = 140 Btu. This tallies with the result given by area measurement. In this cycle there is no need to measure areas. The work done can be found direct by finding the heat drop along the vertical expansion line C D.

As the volume at D is 24 cu. ft. it follows that the piston will have travelled 24 ft. up the cylinder. (The volume of the original water is so small that it can be virtually ignored.) Our string of weights will be 24 ft. – 4 ft. 5 in. = 19 ft. 7 in. long.

80. SUPERHEATED WORKING. Suppose the exhaust steam was to be used for heating purposes. It would be desirable for the exhaust to be quite dry. In order to attain this it will be necessary to superheat the steam and use a cycle like that shown in Fig. 20. A D is the exhaust line at atmospheric pressure. Point D, the exhaust state point, is on the sturation line and is therefore the state of dry steam at 212° F. Now draw line D C′ vertically up from point D until it cuts the 100 psi.a. line at C′. Point C′ must be the desired initial state point such that steam expanded from this state to atmospheric pressure by doing work will be dry at exhaust. The temperature can be read off the diagrams as 600° F. It has a total heat of 1,329 Btu (this can either be read off the diagram or found in the superheated steam table). Steam at point D has a total heat of 1,150 Btu. The original water contained 180 Btu, so that the heat input was 1,329 – 180 = 1,149 Btu. The heat drop was 1,329 – 1,150 = 179 Btu. This gives a cycle efficiency of 15·6 per cent.

The improved efficiency over the cycle in Fig. 19 is due to the fact that the newly added heat strip C C′ c′ c is high and narrow. In other words the heat has been added with relatively little spread.

In reciprocating engines there is a much greater practical gain by superheating than the gain shown in Fig. 20. When the exhaust valve is open the temperature of the steam in the cylinder is the saturation temperature of steam at the exhaust pressure. During the exhaust stroke the metal of the cylinder parts with heat to the exhausting steam. When the inlet or admission valve opens and fresh hot steam enters the cylinder, the first thing that happens is that the cylinder has to be heated up to the temperature of the new steam. If the steam is saturated it follows that condensation must take place and the amount of this initial condensation can be very large indeed. It is not at all unusual for half the steam to be condensed by the cylinder head, walls and piston before it has had a chance to do any work at all. Sixty years ago Professor W. Ripper carried out some classic experiments on a small reciprocating engine. Using fairly highly superheated steam the improvement in efficiency actually achieved on the engine was about twice as great as the benefits to be expected due to the greater heat drop (see Section 246 for Ripper's results).

The reasons for this are that superheated steam is a dry gas and parts with its heat more reluctantly than does saturated steam ; and that if there is no moisture in the cylinder at exhaust there is no flash steam produced which will draw heat from the cylinder walls to provide the latent heat for re-evaporation.

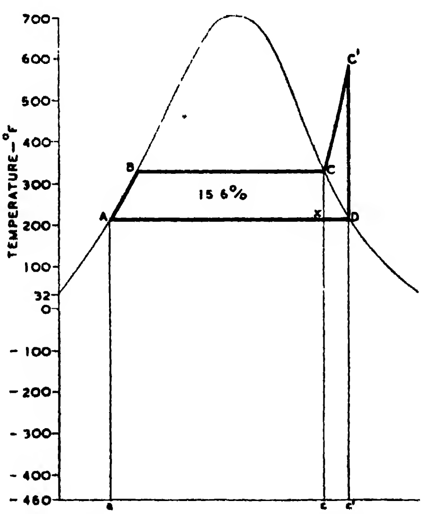


FIG. 20. SUPERHEATED WORKING

81. EXHAUST FOR PROCESS. From an examination of Figs. 4, 7, 8, 16 and 19 it is clear that when heat is added or taken away as *heat* it is shown on the diagram as vertical strips of energy. When work is done it is shown as a piece sliced horizontally off the top of the diagram. What can be done to improve the thickness of the horizontal work-energy slice ?

If we use the exhaust steam at point D, Fig. 20, for heating, say a building, by passing the steam through a set of radiators or heating coils, the steam will condense and give up all its latent heat to the air of the building. The latent heat is area A D c′ a = 970 Btu. The condensate at 212° F. point A will be returned to the boiler for reheating and re-evaporation. In this case the cycle will be 100 per cent. efficient.

Heat input : Sensible 118 + Latent 889 + Superheat 142 = 1,149.
Heat used : Work 179 + Heating 970 = 1,149.

So by using a heating process with the exhaust steam from a power process we can usefully employ the whole of the heat input. The technique of combining power production with process heating is discussed in the next chapter.

82. LOW PRESSURE EXHAUST. Although the use of the exhaust for heating turned the cycle shown in Fig. 20 into a cycle of 100 per cent. overall efficiency, the amount of power produced was only 15·6 per cent. There may not be a heating process available, or it may be necessary to produce as much power as possible before using the steam for heating. It is therefore important to see what can be done to make as much of the energy available as possible for power purposes, by modifying the cycle.

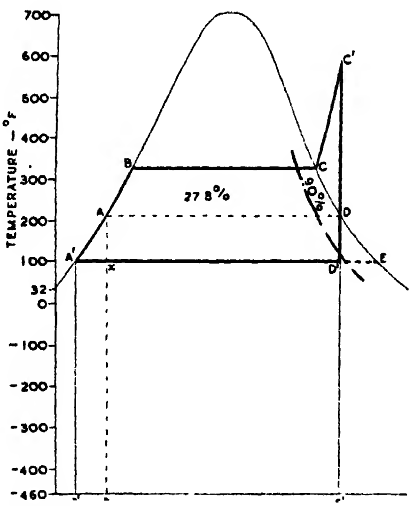


FIG. 21. HIGH VACUUM EXHAUST

Let us abandon our queer experimental cylinder and turn to real engines and turbines. If, instead of the open-ended cylinder, we close in the cylinder and transmit the force exerted on the piston by means of a piston rod passing through a gland in the cylinder cover, we can, by means of a condenser and vacuum pump, relieve the front end of the piston of most of the atmospheric pressure. This also means that the steam behind the piston can be allowed to expand to a pressure well below atmospheric pressure. This will increase the thickness of the horizontal power slice without seriously increasing the spread.

Fig. 21 shows the effect of generating steam at 100 psi.a. superheating it to 600° F., as in Fig. 20, and expanding it down to an exhaust pressure of 28 in. vacuum at 101° F. The total heat at C′ is 1,329 Btu. The total heat at D′ on the 28 in. vacuum line is 978 Btu. This gives a heat drop of 1,329 − 978 = 351 Btu. In order to work at a vacuum the exhaust must pass into a condenser where the heat, area A′ D′ c′ a′ is rejected. The condensate returning to the boiler will be at state A′ at 101° F. and will contain 69 Btu of sensible heat. The total heat input is 1,329 − 69 = 1,260 Btu for a power output of 351 Btu. This shows a cycle efficiency of 27·8 per cent. The gain has been the fine thick power slice A′ A D D′, but we are not getting something for nothing as the extra area A′ A a a′ has been added to the input. A good gain has been got but the spread has been slightly increased by a′ a.

The exhaust at D′ is 11 per cent. wet. This is of little consequence. Steam at 100° F. is not very useful for process heating and 11 per cent. of moisture at exhaust is not objectionable either in an engine or a turbine. The limit of permissible wetness in a turbine exhaust is about 13 per cent. Above this figure the water droplets may wear away the last row of blades.

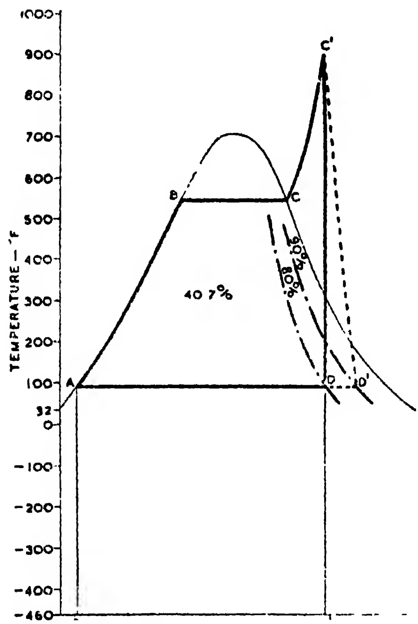


FIG. 22. HIGH PRESSURE AND HIGH VACUUM

83. HIGH PRESSURE WORKING. Inspection of the diagrams shows that there is another useful slice of work to be obtained above the B C line by raising the initial pressure. Let us go right up to the limit of prudence—say 1,000 psi.a. at 900° F. There are a number of stations working at higher pressures and higher temperatures, but it has yet to be conclusively proved that the thermodynamical advantages of such extremes outweigh the practical difficulties of keeping such plants running day-in, night-out.

Fig. 22 shows such a cycle. C′ is the state point of steam at 1,000 psi.a. and 900° F. The total heat at C′ is 1,448 Btu. Point D is near the 28½ in. vacuum line with a temperature of 90° F. and a heat content of 882 Btu. The heat drop is 1,448 − 882 = 566 Btu. The sensible heat in the condensate at A is 58 Btu, so that the heat input is 1,448 − 58 = 1,390. This gives a cycle efficiency of 40·7 per cent.

84. FEED PUMP ENERGY. There is a small additional amount of energy that we have so far neglected. This is the amount of energy needed to force the feed water into the boiler. Except at very high pressures it is so small that it can be neglected except for very exact calculation. In the diagrams given here it would amount to little more than the thickness of the line. The actual effect would be that point B would be just a little bit to the left of the water line. In most ordinary cycles the theoretical energy represented by the feed pump is less than 2 Btu. In practice feed pumps are thermodynamically inefficient and handicap the net power output considerably, especially at high pressures. At the moment we are considering only the theoretical effect of the feed pump on the cycle.

85. EFFICIENCY RATIO. Fig. 22 refers to a perfect engine. Even the largest turbines are far from perfect. Let us assume that the turbine working on the Fig. 22 cycle has an efficiency ratio of 80 per cent. This means that out of a theoretical heat drop of 566 Btu only 453 are actually turned into work. So that the total heat in the exhaust will be 1,448 − 453 = 995 Btu. Now the exhaust pressure is not changed ; the exhaust steam just carries more heat than would the exhaust from a perfect engine. This extra heat is due to steam leaking over the tips of the blades, leaks through the interstage glands, slip and friction in the blades, wiredrawing in the governor valves, etc. So the actual state of the exhaust steam will be at D′ where the 995 total heat line cuts the 90° F. exhaust line. The state point instead of moving vertically will move slightly to the right along some unknown and irregular path from C′ to D′. Now the spread to the right D D′ represents an irreversible process, so that we cannot say that the work done is equal to area A B C C′ D′, which is the area traced out by the state point. It is not. The work done is 80 per cent. of area A B C C′ D. Although we cannot ascertain the work from the areas in such a cycle we can get the real work done from the real heat drop. If we measure the quality of the steam at C′ and at D′ (see Section 213 in Chapter 6) we can find the exact heat drop from which we can calculate the efficiency ratio of the machine.

86. MOISTURE AT EXHAUST—REHEAT. Had the engine in Fig. 22 been perfect the exhaust at D would have been over 20 per cent. wet. This would not have been permissible. With an efficiency ratio of 80 per cent. the exhaust at D′ is only 12 per cent. wet. If the efficiency ratio were as high as 85 per cent. the exhaust would be too wet. If the initial temperature had been 850° F. instead of 900 the exhaust would have been too wet. Where excessive wetness is expected the steam is withdrawn from the turbine at some point during expansion and passed through a separate superheater, built into the boiler, where it is resuperheated. After reheating it is brought back to the

turbine where it can be expanded down to a very low exhaust pressure without any danger of excessive wetness.

The cycle is shown in Fig. 23. Steam at 1,000 psi, superheated to 900° F. is withdrawn from the turbine at about 200 psi and reheated to 800° F. It is then returned to the turbine where it expands down to 28·5 in. vacuum. The cycle efficiency is 41·5 per cent.

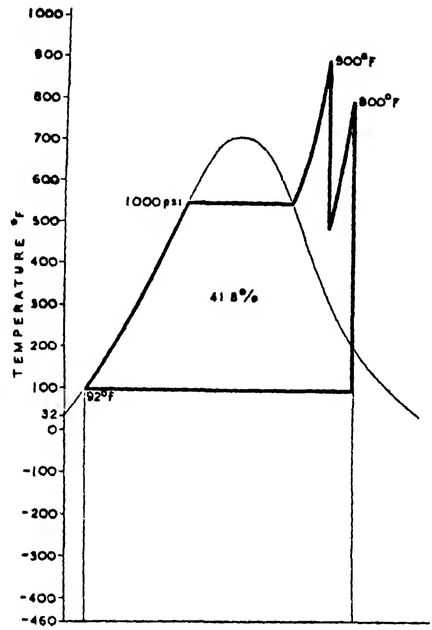


FIG. 23. REHEAT

87. SHORTENING THE ODDS—NARROWING THE SPREAD. It is very clear from Figs. 21, 22 and 23 that the left-hand side of the picture is not pulling its weight. Area A′ A a a′ contains 112 Btu in Fig. 21, but it only yields power equivalent to the poor little triangle A′ A x, which contains 8 Btu—an efficiency of 7 per cent. If we could cut a good vertical slice off the left-hand side of the diagrams we should reduce the spread and yet sacrifice very little of the upper powerful layer.

Condensation takes off vertical slices. If the whole of the exhaust steam is used in a useful condensation process we have seen that the cycle is turned into one of 100 per cent. efficiency. By doing a little condensation process the big power stations can greatly reduce their spread. They actually use multiple pass-out turbines (*see* the next chapter) but the pass-out quantities are relatively small. These turbines are called bleeder turbines and the cycle works as follows.

88. THE REGENERATIVE CYCLE. Fig. 24 shows a modern power station cycle. Initial steam conditions are the same as in Fig. 22, namely, 1,000 psi.a. at 900° F. The steam is put into the turbine at C′ and expands.

At pressure E, about 80 psi, a portion of the steam is bled off, or passed out of the turbine. The amount of heat in this steam is shown diagramatically as the vertical strip E e c′. This steam is taken into a feed water heater and condensed, the condensate being returned to the feed water. This has the effect of adding the heat area P Q q a to the feed water. Further along the turbine, at about 10 psi, another lot of steam is bled off the turbine. The heat in this is shown as strip F f e, which is in turn transferred to the feed (before it receives area P Q q a) as area Q R r q. Finally, a third amount of steam is bled off and used in the first of the series of feed heaters to put in G g f as A′ R r a. By thus stealing a leaf out of industry's book, the big power station, by ingeniously combining a little process heating with power generation has reduced the spread of the cycle by a′ a, has saved the heat loss in area A′ A a a′ while only forfeiting the small power area A′ P A.

Total Heat Drop ..	.. C′—D
Power Forfeit by Bleeds	.. P—A
Net Available Power ..	.. (C′—D)—(P—A)
Heat Input	.. C′—P

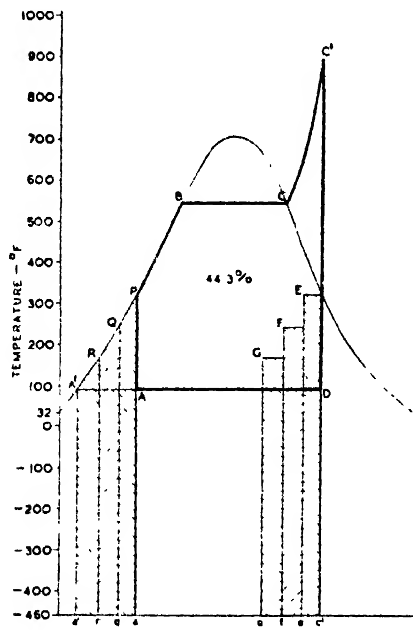


FIG. 24. REGENERATIVE CYCLE

This gives the cycle an efficiency of 44·3 per cent. If the turbine efficiency ratio is 82 per cent., the generator efficiency 98 per cent., the boiler efficiency 85 per cent., and if the auxiliaries take 4 per cent., the overall coal/electricity efficiency will be 29·1 per cent., which tallies with the performance obtained by the fairly good stations.

A method of cutting an even bigger strip off the left side of the diagram has been suggested. This Compression-Regeneration cycle is described in the Appendix, Section 803.

89. REGENERATIVE REHEAT CYCLE. It is possible to combine the regenerative and the reheat cycles, and this is in fact done in some stations. Such a cycle is shown in Fig. 25, where the cycle efficiency is now 46·3 per cent. By working at pressures higher than 1,000 psi, by improving the vacuum to 29 in. and by using rather more superheat, it is possible to squeeze the cycle efficiency up to just over 50 per cent., when the actual efficiency that can be attained is about 31 per cent.

90. SPREAD. From all the foregoing cycles it is proved that every improvement in efficiency has been got by decreasing the spread compared to the total heat input. It should also now be clear that, while an increase in spread sometimes means an increase in the amount of energy available for power production, it ALWAYS means an increase in the amount of heat that must be rejected in the exhaust.

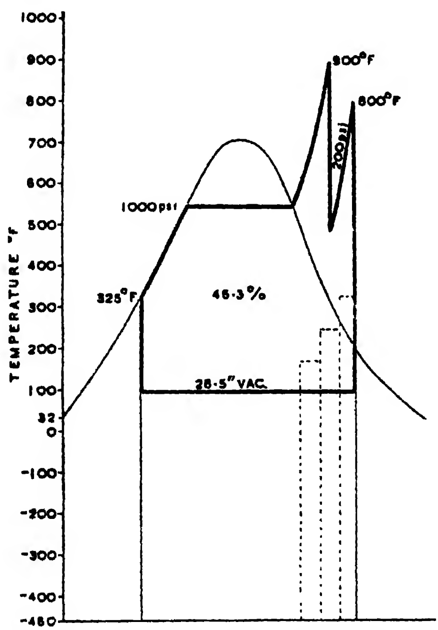


FIG. 25. REHEAT AND REGENERATION

If we have a certain quantity of heat with which to work a steam engine cycle for power generation, we can liken it to a pint of whisky. We could pour the whole pint down one man's throat and, although he might enjoy it very much at first, he would not enjoy it long ; he would suffer a breakdown ; his availability would be low. We could, in the same way, use our heat to generate a small amount of very high pressure, very high temperature steam, and a large

proportion of the heat would theoretically be available for power production. Such steam would quickly wreck any engine or turbine we could build. On the other hand, we could dilute the pint of whisky and so spread it out that it was no use to anyone. Similarly, we could use our heat to raise a very large quantity of steam at 28 in. vacuum, but it would be useless for generating power. We must, therefore, choose some point between these extremes to suit practical working conditions. It might be four fingers and a splash—1,200 psi and reheating with stage bleeding— ; or one finger and half a siphon—250 psi and 150° F. superheat. While it is important to keep the spread to a minimum during the raising of steam, it is even more important to prevent any increase of spread in the actual power conversion part of the cycle—such an increase is dead loss It is, however, sometimes permissible to increase the spread in some heating processes—but that is another story.

91. CARNOT'S THEOREM. In 1824 Sadi Carnot, a French physicist, propounded the fundamentals of the perfect heat engine cycle. Fig. 26 shows the Carnot cycle.

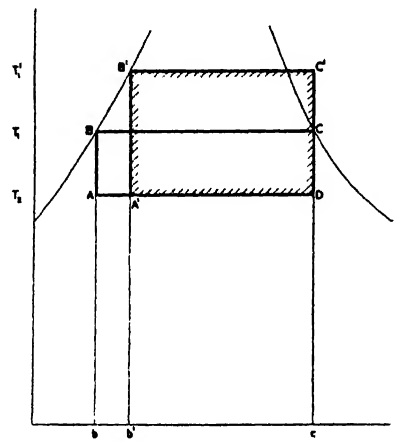


FIG. 26. CARNOT'S CYCLE

A B is perfect compression. This is comparable in the working steam cycle with getting the water into the boiler and raising it to boiling temperature.

B C is heat addition at constant high temperature. This part of the cycle is comparable with evaporation of the water and adding heat to the steam in a working cycle.

C D is perfect expansion where all the heat energy liberated is converted into useful work.

D A is heat rejection at low temperature.

The total energy supplied is area B C c b and the energy available for doing work is area A B C D.

As these areas are both rectangles on a common base, we can say that the total energy supplied is proportional to the temperature T_1 during heat addition.

Therefore the energy available for doing work is proportional to the temperature drop across the cycle, or $T_1 - T_2$.

Now the efficiency of a cycle is expressed by dividing the useful work by the heat input.

In Fig. 26 this efficiency is clearly $\frac{T_1 - T_2}{T_1}$.

So we now see that instead of having to accept Carnot's theorem parrot-like, it is quite obvious and logical.

Fig. 26 shows that by increasing the T_1 temperature to T_1' we reduce the spread from b c to b′ c ; we increase the output by C C′, but we require superheat at C′.

All the improvements that we have made in progressing from Fig. 16 to Fig. 24 (apart from superheating which is done principally to limit condensation for practical reasons) have tended to make the diagram approach a tall narrow Carnot rectangle.

92. WHAT IS SPREAD? Rankine and Clausius emphasised the importance of keeping the spread small and limiting its increase. Rankine called spread the " Thermodynamical Function ". Fortunately this name did not catch on. Clausius called it ENTROPY. Unfortunately this name stuck, and has been a stumbling block ever since. The dictionaries say that " entropy " is derived from the Greek ἐντροπή and that it means a " turning-in ". That is just what it does not mean ; it means a spreading-out. But Clausius may have had his tongue in his cheek and may have intended entropy to have its Modern Greek meaning of " Shame."

93. ENTROPY. Entropy clearly means spread in the energy diagrams we have been considering. These diagrams are called Temperature-Entropy diagrams. In some dictionaries the meaning of entropy is simply given as " the length on a diagram whose area is heat and whose height is absolute temperature ". (A remarkable case of passing the buck.) Other dictionaries define entropy as an " evening-out " or as a " degeneration " (these are notoriously synonymous.) Does entropy always mean spread ? The astronomer tells us that the entropy of the sun is continually increasing. We can reply that it is obvious that the sun is continually spreading its energy through the celestial void. The geologist tells us that the entropy of the earth's crust is increasing. We can retort that we know well that rains and frosts are spreading the mountains over the valleys and plains, and that the rivers are spreading the land over the sea bed. The statistician tells us that a system has maximum entropy when its components are in random distribution. We can say that when our wife makes a good cake the currants are spread higgledy-piggledy throughout the cake. So—we can take it that entropy means spread and, as the diagrams have shown, any increase of entropy, or " spreading of energy " lowers the availability of that energy for doing useful work.

94. ENTROPY ALWAYS INCREASES. On the diagrams it has been shown that by doing a heating process—that is by taking heat from our steam—the entropy of the steam is reduced. The heated substance, on the other hand, must have a lower temperature than the heating steam so that there must be a

larger increase of entropy in the heated material than there was a reduction of entropy in the heating steam. This must be true. The heat in a heated material must be more spread, less concentrated, than in the heating medium. So heat transfer must always be accompanied by an overall increase of entropy (except when carried out in a perfect counter current heat exchanger between materials of the same weight and the same specific heat).

It has been shown that in a perfect engine power would be generated by steam without any increase of entropy. What happens to the power ? It is all eventually spread out and dissipated at low temperature. The power may go into an electric radiator. The energy spreads itself out as low grade heat in the manager's office ; it increases the entropy of his sanctum. Take an ocean liner. Energy is liberated at very high temperature and low entropy in the boiler furnace. Twenty per cent. of this valuable energy is lost up the funnel and ventilators. There is a small increase in the temperature and entropy of the atmosphere. Of the 80 per cent. energy going to the turbines, 50 is thrown away in the condensers and causes a small rise in oceanic temperature and entropy. The remaining 30 per cent. is converted with only a small increase of entropy into useful work at the propeller. All the energy at the propeller, however, is converted back into very low-grade heat—slip in the propeller—blade friction—the sideways displacement of the water by the ship's bows—the making of waves—the friction of the ship's skin—the air resistance of the superstructure. So that in crossing the Atlantic all we have done is to spread all the good energy in the oil or coal that was burnt over 3,000 miles of sea and air, as an irrecoverable tiny rise of temperature accompanied by a corresponding increase of oceanic and atmospheric entropy. (We got to America, of course.)

Although condensation is a process which reduces the entropy of the condensing steam there is an even greater increase in the entropy of the cooling medium, be it atmosphere or river. Half of the energy liberated in the boiler furnaces of London's riverside power stations is spread uselessly into the river Thames, which is theoretically increased in temperature by more than 10° F. So that if a system is looked at as a whole, the past can always be distinguished from the present or future by an increase of entropy as time proceeds. It is because entropy is always increasing throughout the universe that an eminent physicist has described entropy as " Time's Arrow ".

95. THE NUMERICAL VALUE OF ENTROPY. In what units is the entropy of steam measured ? An area is height multiplied by breadth. So that breadth, or spread, is area—Btu—divided by absolute temperature. This means that the entropy scale is in units of Btu per ° F. (Does this ring a mental bell ? Specific heat is also Btu per ° F. So that an increase of entropy can be looked on as a kind of rise in specific heat. Some people find this a helpful conception.)

Take point Z in Fig. 27.

It is on the 32° F. line and on the 1,000 Btu line.

The area A Z z a = 1,000 Btu and the height a A = 492° F. abs.

Therefore the spread, or entropy, is 2·033 Btu/° F.

Now take point Y where the 500 Btu line cuts the 32° F. line.

The height is again a A = 492° F. abs.

The area A Y y a is 500 Btu, so that the entropy is 1·016 Btu/° F.

Take point X on the 1,000 Btu line at 500° F.

If 1,000 is divided by 500 + 460 = 960, the result is 1·042 when in fact the entropy on the diagram is 1·222.

Why the discrepancy ? The area representing the heat at point X is not a rectangle, but is area A B X x a. This discrepancy is of no consequence as we are never concerned with the absolute value of entropy, only with changes of entropy or with processes carried out with no change of entropy.

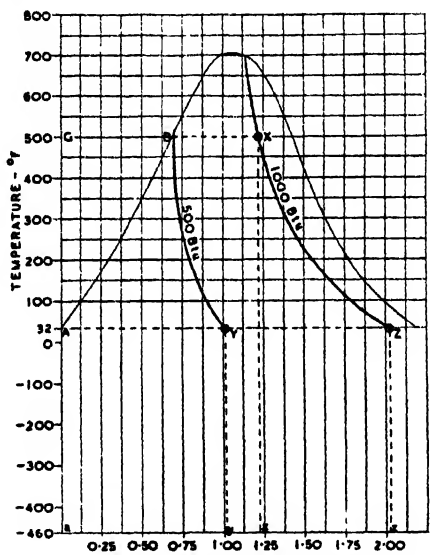


FIG. 27. ENTROPY UNITS

96. GIBBS' FUNCTION. If we calculate the total heat at point X in Fig. 27 by multiplying the absolute temperature by the entropy, we shall get too high a result due to the inclusion of the triangular area A G B. If we knew the value of this triangle for any pressure it would simplify some of our calculations. For this reason it is included in many steam tables and is called " G " or Gibbs' Function (after its inventor in 1875, Willard Gibbs), or sometimes it is graced by the terrific title of " Negative Thermodynamic Potential ". As will be seen later its use provides a valuable short cut. The values of Gibbs' Function are given in Table I, the Saturated Steam Table.

97. ADIABATIC OPERATIONS. When steam expands or is compressed in a reversible manner without gaining or losing heat through the walls of the machine, the expansion or compression is said to be " Adiabatic ". If entropy is a bad word, " adiabatic " is even worse. It is derived from the Greek ἀ διά βαίνειν and therefore means " not passing through ". Clearly no heat passes through during expansion or wiredrawing through a reducing valve or

orifice, but engineers use the word adiabatic to distinguish mechanical expansion from wiredrawing. When adiabatic expansion or compression is spoken of, what is meant is perfect mechanical compression or perfect mechanical power-producing expansion. In other words, expansion or compression at constant entropy. It would be much less confusing were such operations called " ideal ", or even " Isentropic " expansions and compressions. The use of the word adiabatic could then be reserved for such phrases as : " I bought a fine adiabatic mackintosh to-day ", or " The inner tubes on my bicycle are no longer adiabatic ".

98. ADIABATIC HEAT DROP. The heat drops from C or C′ to D on Figs. 19, 20, 21 and 22 are the amounts of energy available for conversion into work by a perfect engine. They take place at constant entropy and are called "Adiabatic Heat Drops ".

99. THE PRACTICAL USE OF ENTROPY VALUES. The steam tables give figures for entropy for all the states of steam and water. By using these values and doing a little simple arithmetic we can calculate the heat drop, wetness, volume, etc., without trying to read them off the diagram. But we want a rough diagram to help to keep us on the right lines. A diagram sufficiently large to be accurate to within 1 or 2 Btu is very large and cumbersome, whereas the steam table is portable and can be used anywhere without inconvenience. Let us confirm by calculation some of the results we have obtained by measurement on the diagrams.

100. TO FIND ALL QUALITIES AT EXHAUST. In Fig. 19 we had saturated steam at 100 psi.a. which we propose to expand adiabatically (we must conform to conventional nomenclature) down to atmospheric pressure, from point C to point D. We want to know the heat drop from C to D, the wetness at D and the volume at D. Clearly the length of line A D compared to the total length A E of the atmospheric evaporation line represents the amount of dry steam at point D. Having found the percentage of dry steam we can then find the total heat in the mixture and the volume of the mixture. The entropy at D will give us the wetness at D, and the entropy at D is the same as the entropy at C.

The saturated steam table gives the following qualities for 100 psi.a. steam at point C :—

Pressure	100 psi.a.
Total heat	1,188 Btu
Total entropy	1·604 Btu/° F.

From the steam table we also get the qualities of saturated steam at the exhaust—atmospheric—pressure :—

Sensible heat	180 Btu
Latent heat	971 Btu
Total entropy	1·757
Evaporation entropy	1·445
Water volume	·0167 cu. ft.
Steam volume	26·8 cu. ft.

The reduction D E in evaporation entropy in Fig. 19 due to wetness

$$= 1{\cdot}757 - 1{\cdot}604 = {\cdot}153.$$

The entropy of evaporation of 100 per cent. dry atmospheric steam A E

$$= 1{\cdot}445.$$

The wetness at state D

$$= \frac{{\cdot}153}{1{\cdot}445} = {\cdot}1059 \text{ lb. water.}$$

$$= \underline{10{\cdot}59 \text{ per cent. wet.}}$$

The total heat at D is made up of the sensible heat in 1 lb. of water at 212° F. plus the latent heat in (1 − ·1059) lb. steam

$$= 180 + (1 - {\cdot}1059)\ 971.$$
$$= 1{,}048 \text{ Btu.}$$

The adiabatic heat drop $= 1{,}188 - 1{,}048 = \underline{140 \text{ Btu.}}$

The volume of the mixture at D is the volume of ·1059 lb. water plus the volume of (1 − ·1059) lb. of saturated steam at atmospheric pressure

$$= ({\cdot}1059 \times {\cdot}0167) + (1 - {\cdot}1059)\ 26{\cdot}8.$$
$$= {\cdot}0018 + 23{\cdot}962.$$
$$= \underline{23{\cdot}964 \text{ cu. ft.}}$$

Clearly the volume of the water can always be ignored unless the wetness is very large or the pressure is very high.

101. TO FIND THE HEAT DROP USING GIBBS' FUNCTION. If we only wish to find the heat drop and if we have a table giving the values of "G" the calculation is very short. There is no need to calculate the wetness first. The total heat at D will be the absolute temperature multiplied by the entropy less the value of the missing Gibbs' triangle "G". The value of G at atmospheric pressure is 30 Btu, so we have :—

Total heat at D $= [(212 + 460)\ 1{\cdot}604] - 30 = 1048.$

Heat drop $= 1{,}188 - 1{,}048 = \underline{140 \text{ Btu.}}$

102. TO FIND THE NECESSARY SUPERHEAT. In Fig. 20 it was postulated that steam at 100 psi.a. was to be so superheated at C′ as to be dry at exhaust D. We wish to determine the amount of superheat necessary at C′.

The entropy at C′ must be the same as the entropy at D, which is the total entropy of dry saturated steam at atmospheric pressure, which the steam table tells us is 1·757.

This then must be the entropy of the superheated steam at 100 psi.a.

Inspection of the superheated steam tables shows that at 100 psi.a. the entropy is 1·758 at 600° F. and 1·749 at 580° F.

We can therefore say that 1·757 entropy corresponds at 100 psi.a. to a steam temperature of <u>598° F.</u> (We had read this temperature off the diagram as 600° F.)

103. TO FIND THE POWER AVAILABLE. Consider the case of an exhaust turbine taking steam exhausted from a reciprocating engine at atmospheric pressure. The engine exhaust is 4 per cent. wet, and the amount of steam passing is 20,000 lb. per hour. What power can be expected from the turbine when exhausting into a condenser at 28 in. vacuum ?

The conditions at the turbine inlet, which are those of the engine exhaust are found in the steam table for saturated atmospheric pressure steam :—

Liquid entropy	·312
Evaporation entropy	1·445
Sensible heat	180
Latent heat	970

The engine exhaust is 4 per cent. wet, and so has a dryness fraction of ·96.

The entropy of the engine exhaust will be :—

$$\cdot 312 + (\cdot 96 \times 1\cdot 445) = 1\cdot 699.$$

The total heat in the engine exhaust and at the turbine inlet is :—

$$180 + (\cdot 96 \times 970) \quad = 1{,}111.$$

Now, knowing the initial entropy we could calculate the total heat in the turbine exhaust by means of Gibbs' Function, but as we must find out the amount of wetness in the turbine exhaust anyhow, nothing would be saved.

The steam table gives the following qualities for saturated steam at 28 in. vacuum :—

Total entropy	1·980
Evaporation entropy	1·849
Sensible heat	69
Latent heat	1,037

If the steam leaving the turbine were dry saturated it would have an entropy of 1·980, but we know that the initial entropy is 1·699. The difference represents moisture in the steam.

Difference in entropy = 1·980 − 1·699 = ·281 which is the entropy due to exhaust moisture.

The evaporation entropy at 28 in. is 1·849 so that the wetness at the turbine exhaust is

$$\frac{\cdot 281}{1\cdot 849} = \cdot 152 \text{ or } 84\cdot 8 \text{ per cent. dry.}$$

The total heat in the turbine exhaust is $69 + (\cdot 848 \times 1037) = 948$ Btu.

The heat drop with adiabatic expansion is $1{,}111 - 948 = 163$ Btu.

One horse-power is equivalent to 1,980,000 ft. lb./hour and 1 Btu is equivalent to 778 ft. lb. *See* Table VIII, Section 53.

The power available is $\dfrac{20{,}000 \times 163 \times 778}{1{,}980{,}000} = \underline{1{,}281 \text{ H.P.}}$

104. EFFICIENCY RATIO. The foregoing has assumed a perfect engine or turbine. As explained in Section 85 losses occur. It is difficult to predict in general terms just how great these losses will be. An approximate idea of

just what order of efficiency can reasonably be expected from turbines is given in Table IX. This table requires explanation ; it must be used with reserve, and it must be used correctly. Because this table and Table X suggest a probable efficiency ratio, it does not follow that this will be forthcoming in practice ; in some cases the tabled figure may not be reached ; in other cases it may be exceeded. Engine and turbine builders must use patterns that are nearest to requirements, and such patterns may perhaps not be very suitable, or may indeed exactly fill the bill.

The efficiency ratio of an engine or turbine is the percentage of the ideal or adiabatic power in a cycle that is actually converted by the machine into useful work. Suppose the efficiency of a cycle is 30 per cent. and that the efficiency ratio is 50 per cent., then the engine will have an actual thermal efficiency of 15 per cent.

To enable Table IX and its uses to be understood, there is an essential point regarding engines and turbines that must be understood. A turbine or engine operating over a small pressure drop will, for a given power, be a much larger machine than one operating over a large pressure drop—much more steam will pass through it. Again, a machine operating over a particular pressure or heat drop at low pressure will be a much larger machine than one working over the same heat drop at high pressure. The percentage losses in engines and turbines decrease as the volume of the machine increases (within limits). This is only to be expected. In a reciprocating engine the larger the cylinder the less is the proportional cooling effect of the cylinder walls and the less is the relative area of possible leaks. In a turbine the larger the machine the smaller are the clearance spaces relative to the effective blade area. The efficiency of a prime mover depends to a great extent on its volume—the larger the volume the less the proportionate losses.

TABLE IX. APPROXIMATE TURBINE EFFICIENCY RATIOS

(Assuming appropriate Superheat and appropriate Speed)

Pressure Drop \ Horse Power	250	500	750	1,000	2,000	3,000	4,000	5,000	7,500	10,000
1,000 psi.a. to 400	–	—	—	—	59	62	64	65	67	68
400 psi.a. to 100	47	54	57	59	64	67	68	69	71	72
100 psi.a. to 20 ..	61	65	67	68	71	73	74	75	76	76
20 psi.a. to 28″ vac.	64	68	70	71	74	76	78	79	80	80

(This table must not be used without reading the explanation in the text.)

In Table IX four pressure drops are given. Each of these pressure drops represents roughly similar amounts of available energy. We can see how the efficiency rises with increase of volume. Take the second line—initial pressure 400 psi, exhaust pressure 100 psi. We see that a 250 H.P. turbine can be

expected to have an efficiency ratio of 47 per cent. As the size of the machine rises so does the efficiency increase until at 10,000 H.P. we can expect the efficiency ratio of 72 per cent. Now take the 2,000 H.P. column. If the machine operates between 1,000 psi and 400 psi the expected efficiency ratio is only 59 per cent. At these high pressures the steam occupies a very small volume and the turbine is consequently small with proportionally big clearances. Steam at these pressures is also dense and there is considerable friction between the blades and the steam. As the volume of the steam rises so does the volume of the turbine increase and the efficiency ratio improves until at the low pressure range the efficiency ratio that can be expected is 74 per cent.

In using the table it must be understood that the horse power refers to only one pressure drop at a time. Some examples will make this clear. Suppose we wish to generate 2,000 H.P. The figures under 2,000 H.P. show the efficiency ratios that can be expected if all the 2,000 H.P. is generated over one pressure drop. That is to say, a 2,000 H.P. turbine taking steam at 400 psi and exhausting at 100 psi can be expected to have an efficiency ratio of 64 per cent. Now suppose we wish to generate 2,000 H.P. between 400 psi and 20 psi. This must be looked upon as two 1,000 H.P. machines operating under two separate pressure drops. The 1,000 H.P. column shows that we can expect 59 per cent. and 68 per cent. from them. So that although much less steam will be used the efficiency ratio will be no more than before because the machine is really a smaller machine—it must be, because it is taking much less steam. Now suppose it is suggested that 2,000 H.P. be generated using an initial pressure of 1,000 psi exhausting into a condenser at 28 in. vacuum, this must be looked upon as four 500-H.P. machines. A 500-H.P. turbine working between 1,000 and 400 psi is so small that it will probably not pay to use it alone, although its shortcomings would be to some extent offset if it were, as in this case, merely the small end of a big machine.

Table IX assumes good conditions ; adequate superheat and proper turbine speeds. It will, for example, not be likely that an efficiency ratio of 68 per cent. could be obtained from a 1,000-H.P. machine operating between 100 and 20 psi.a. if the steam is saturated and if the speed is only 1,500 r.p.m.

Section 80 has shown that an increased cycle efficiency can be obtained by superheating the steam, apart from the beneficial effect of superheating by preventing condensation. Now the hotter steam is, for a given pressure, the larger the volume it occupies. So that the use of superheat not only improves the cycle efficiency, it improves the efficiency ratio also.

A much greater gain can be expected from a lowering of the back pressure than from raising the inlet pressure. Figs. 20, 21 and 22 explain the reasons for a higher cycle efficiency, while Table IX and the explanations just given show that the efficiency ratio will also be increased. There are other arguments against the use of a very high initial pressure, and in favour of a low back pressure. High pressures call for high superheat which introduce engineering difficulties. If the prime mover is exhausting into a process main it will be found beneficial to work the process at the lowest possible pressure. This has been referred to in Chapter 1 and will frequently be spoken of again. High boiler pressures call for big power for the feed pumps and instead of the boiler

feed pumps representing a trifling debit against the power generation credit, it may become a serious drain. (There is a factory in England working at extremely high pressure where the boiler feed pumps take 10 per cent. of the total power generated.)

The turbine discussed in Section 103 should, according to Table IX, have an efficiency ratio of about 72 per cent.

The actual H.P. developed would therefore be 72 per cent. of 1,281 or 922 H.P.

The real effective heat drop will be 72 per cent. of 163 = 117 Btu.

So that the total heat in the turbine exhaust will be 1,111 − 117 = 994 Btu.

The real dryness of the steam will be $\frac{994 - 69}{1{,}037} = \cdot 89$ or 89 per cent.

The wetness in the turbine exhaust will be 11 per cent. not 15·3 per cent. and can safely be permitted.

Table IX shows probable efficiency ratios for turbines only. Reciprocating engines are not so straightforward. Table X shows probable efficiency ratios in a different form. It shows the figure that can be expected from a machine taking 10,000 lb. of steam an hour under various conditions of inlet and exhaust pressure for both engines and turbines. It assumes saturated steam is used in all cases. These conditions have been selected as they are likely to apply to thousands of small and medium sized factories. The first figure in each of the triple entries in this table is the adiabatic or ideal horse power.

Let us take conditions when the machine is exhausting to atmosphere, by running our eye down the ATM column. It will be seen that the efficiency ratio of the engine drops as the pressure across the engine increases, because there will be a greater range of temperature in the cylinder and consequently more cylinder condensation. The turbine, however, shows a small increase in efficiency ratio because, as temperatures remain constant in any one part of a turbine, there is no equivalent to cylinder condensation, and, as the machine gets bigger, the efficiency rises.

Now let us see the effect of a constant inlet pressure with various exhaust pressures. Take the 200 psi line. It will be seen that the ideal H.P. decreases as the exhaust pressure goes up ; that is to say, the machine becomes smaller the higher the back pressure. It follows that the turbine should show a steadily decreasing efficiency ratio with rising exhaust pressure, and this the table shows. At 20-in. vacuum exhaust the ideal H.P. is 1,020 and the turbine efficiency ratio is 62 per cent., so that we can expect to get 632 H.P. At 60 psi.g. back pressure the ideal H.P. is only 330 and the turbine efficiency ratio is only 39 per cent., so we cannot hope to get more than 129 B.H.P. The engine, however, shows a steady increase of efficiency ratio up to a moderate back pressure. This is due to two causes ; first an engine cannot take full advantage of vacuum exhaust because the low pressure cylinder would be too large for practical working ; second, because of the smaller temperature changes in the cylinder at the higher back pressures, there is less cylinder condensation. As the back pressure rises beyond 20 psi.g. the engine becomes such a small unit that its efficiency ratio starts to drop.

General inspection of the table shows how well a turbine can use very low pressure steam and what good use it can make of vacuum exhaust. It also shows how ill-adapted is a small turbine to work on high pressures. It is interesting to note that at 10-in. vacuum and 225 psi inlet, both the engine and the turbine can be expected to have the same efficiency ratio of 56 per cent. and should yield 510 H.P. from 10,000 lb. of saturated steam an hour.

TABLE X. IDEAL HORSE POWER AND APPROXIMATE EFFICIENCY RATIOS FOR ENGINES AND TURBINES USING 10,000 LB. SATURATED STEAM PER HOUR

INLET PRESS. SAT. PSI.G.		EXHAUST											
		VACUUM					GAUGE PRESSURE						
		20"	15"	10"	5"	ATM.	5	10	20	40	60	80	100
100	HP	855	760	690	635	590	475	455	355	230			
	E	53	58	60	62	65	64	61	60	56	—	—	—
	T	67	62	60	57	53	51	48	46	37	—	—	—
125	HP	910	815	745	680	650	570	520	420	290			
	E	53	58	60	62	64	67	64	63	60	—	—	—
	T	64	60	59	57	53	52	48	47	38	—	—	—
150	HP	955	860	810	735	695	615	560	465	340	250		
	E	53	57	58	62	63	66	66	66	62	58	—	—
	T	63	59	57	57	53	53	50	48	41	32	—	—
175	HP	985	895	852	775	735	655	595	500	380	290	215	
	E	53	57	58	61	62	66	67	67	62	62	60	—
	T	62	59	56	56	53	53	50	48	43	36	—	—
200	HP	1,020	930	885	810	770	690	635	540	420	330	255	195
	E	53	56	57	60	61	64	66	67	65	62	61	60
	T	62	59	56	56	53	53	50	49	45	39	35	—
225	HP	1,050	960	910	840	800	725	665	575	450	360	290	230
	E	52	55	56	59	60	63	65	66	67	63	62	60
	T	62	58	56	56	54	53	50	49	45	39	36	—
250	HP	1,075	985	940	870	825	750	700	610	485	395	325	265
	E	—	—	—	—	59	62	63	65	67	64	61	60
	T	62	58	56	56	54	53	50	49	46	41	37	—
300	HP	1,120	1,030	980	910	870	795	740	650	530	445	375	315
	E	—	—	—	—	—	—	62	64	67	64	60	60
	T	60	58	56	56	54	54	52	50	46	41	37	—
350	HP	1,150	1,065	1,015	945	910	835	780	690	575	485	415	360
	E	—	—	—	—	—	—	62	64	66	64	60	60
	T	60	59	57	56	54	54	53	50	46	4	38	—
400	HP	1,190	1,100	1,055	980	945	870	815	730	615	525	460	405
	E	—	—	—	—	—	—	—	63	65	63	59	58
	T	60	59	57	56	54	54	52	50	46	44	39	35

(The fact that Tables IX and X do not quite agree is because they are based upon different postulates, and were also based on data received from different sources. The difference between " size " in the two tables must be remembered.

In Table IX the "size" is the output in H.P. In Table X the "size" is constant at 10,000 lb./hour. The volumes of machines considered by the two tables change for different reasons.)

The reason that small turbines are inefficient (smallness may be due to small steam flow, high pressure, high pressure drop, or any combination of these) is that blade clearances cannot be reduced when the length of blade or the diameter of the rotor is reduced. Short blades are inefficient because the shrouds and roots cause disturbances in the steam flow and with blades of ½-in. or so these disturbances are felt over the whole blade length.

The uniflow engine would show up much better than the figures in Table X because the loss by condensation is so much less than with a normal engine. The hot or inlet end of the uniflow engine is only cooled for a brief moment at exhaust point. During the return stroke any heat lost by the hot end goes into the residual steam which is being compressed, and this heat is thus trapped in the cylinder and not lost. During exhaust in a normal engine the whole cylinder and especially the inlet end is giving up heat throughout the whole exhaust stroke and this heat is all ejected with the exhaust.

105. STEAM POWER DIAGRAMS. It has been shown that it is possible to calculate, from the figures given in the steam tables, heat drops, superheat requirements, exhaust conditions, etc., without reference to any diagram. But the temperature entropy diagram has proved to be an exceedingly useful device to induce clear thinking as to what is actually going on. It has been shown that all the qualities that are listed in the steam tables can be read direct off the temperature entropy diagram. Actually some difficulty may be experienced when this is tried. The total heat lines are curved and are not evenly spaced. The pressure and volume lines are even more tiresome in their spacing. It is difficult therefore to interpolate values correctly. Many different diagrams have been tried, but two are of outstanding merit, the temperature entropy diagram that we have been investigating and the Mollier diagram.

106. THE MOLLIER DIAGRAM. This is a diagram constructed with Btu as one scale and entropy as the other scale. The area of the diagram is meaningless and the diagram is of little value in solving the problems of thermodynamical strategy, but as a tactical weapon it is superb. The diagram is named after its inventor, Dr. R. Mollier. A skeleton example is shown in Fig. 28. (A large three-colour working Mollier chart is published by Edward Arnold & Co., London. The values in the diagram are taken from Callendar's 1939 Steam Table and will show a very slight discrepancy with the values given in Keenan and Keyes' 1936 Tables.)

Both Callendar and Keenan and Keyes give values in their steam tables to five significant figures. There is no divergence of any practical importance between the two tables. Only at very high pressures and temperatures is there marked deviation and this only affects the boiler or turbine designer. In no factory problem is there any need to attempt to get nearer than within about 1 per cent.—it would be a hopeless task anyhow. It follows therefore that for practical purposes it does not matter which steam table is used.

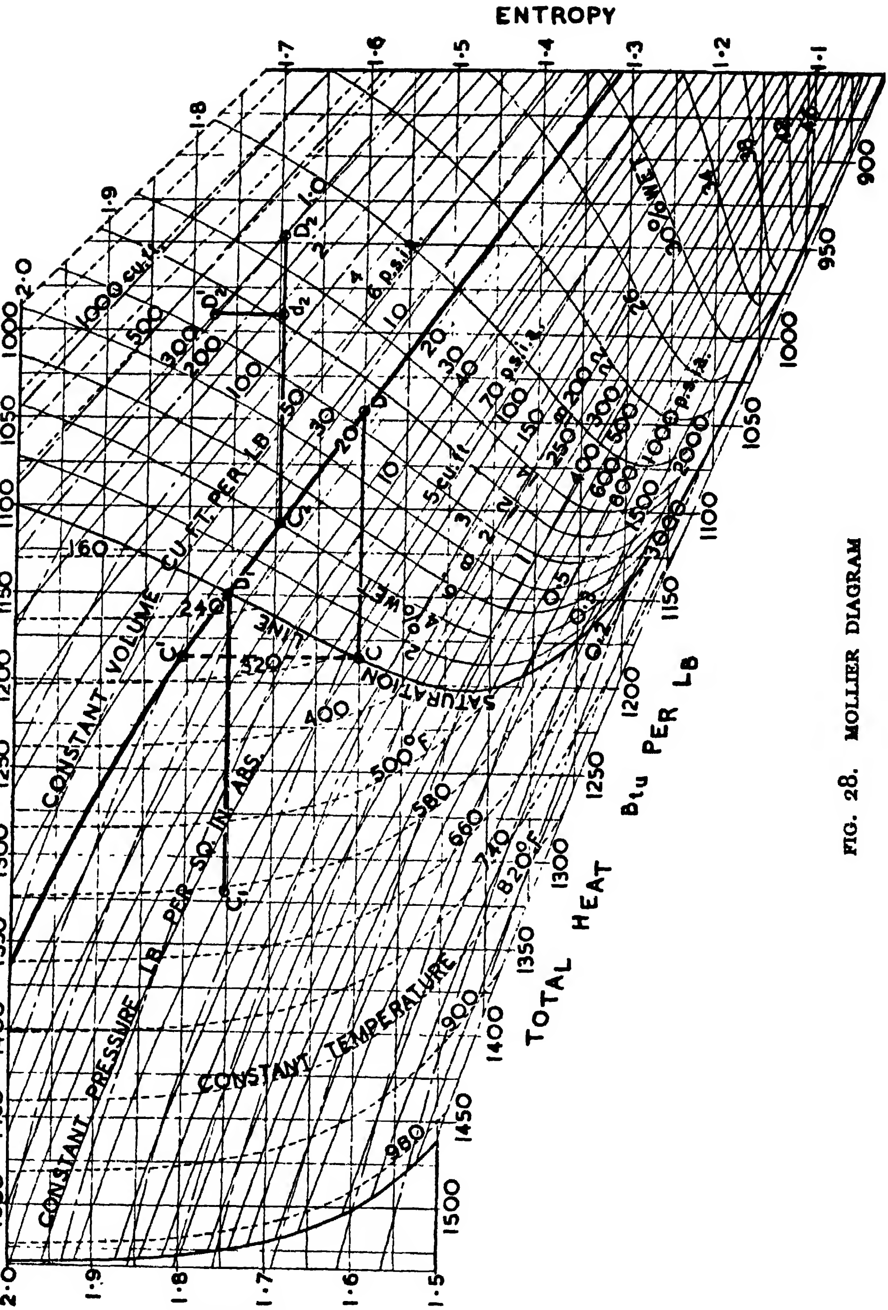


FIG. 28. MOLLIER DIAGRAM

107. THE USE OF THE MOLLIER DIAGRAM. In Section 100 the heat drop, wetness and volume were found for the exhaust condition shown in Fig. 19 by means of the steam table and a little arithmetic. Let us see how easily these answers can be found on the Mollier diagram.

Point C in Fig. 28 shows the state point of saturated steam at 100 psi.a.

Expansion takes place at constant entropy down to atmospheric pressure along line C D.

Length C D is the adiabatic heat drop, 140 Btu.

Point D is 89 per cent. dry and has a volume of 24 cu. ft.

Now, in Fig. 20, the conditions are that the initial superheat at 100 psi.a. must be sufficient to give dry saturated exhaust at atmospheric pressure.

Point D_1 in Fig. 28, on the saturation line at atmospheric pressure represents the desired exhaust state.

Line D_1 C_1 shows the expansion line which cuts the 100 psi.a. pressure line at C_1, where the temperature is 600° F.

The heat drop is length D_1 C_1 and can be directly measured as 179 Btu.

In Sections 103 and 104 it was shown how to find the true heat drop over an exhaust turbine taking atmospheric steam at 4 per cent. wet when the efficiency ratio is 72 per cent. This operation is exceedingly quick and easy on the Mollier diagram.

Point C_2 represents steam at atmospheric pressure 4 per cent. wet.

Adiabatic expansion down to 28 in. vacuum is shown by line C_2 D_2.

The measured length of C_2 D_2 gives a heat drop of 163 Btu. But the efficiency ratio is 72 per cent. so that the actual heat drop is 72 per cent. of C_2 D_2 measured from C_2, or 117 Btu, length C_2 d_2.

The exhaust must contain 994 Btu, the heat content at d_2, and we know it is at 28 in. vacuum.

So point D_2', where 994 is cut by the 28-in. line is the true exhaust point.

In Fig. 17 saturated steam at 100 psi.a. was expanded down to atmospheric pressure by being blown into a larger volume. This is shown on Fig. 28 as line C C′, a vertical constant total heat line. At C′ we can read off the properties of the steam after this wiredrawing ; temperature 290° F., volume 30 cu. ft.

Almost all published Mollier diagrams are drawn for absolute pressures. That is why absolute pressure has been used in this chapter. Any interested person should buy a good Mollier diagram. Apart from the Callendar diagram published by Arnold, there is a good diagram included in the K. & K. Tables. Fig. 28 is too small for practical use. (See Section 805.)

* * *

There are many conflicting requirements in the design of a good engine or turbine, some of which will be discussed further in the next chapter. It is the reconciliation of all these contradictions that constitutes the art of the steam engineer.

* * *

CHAPTER 3

COMBINED POWER AND HEATING

> . . To aid our cause, although we be but two.
> Great is the strength of feeble arms combined.
>
> HOMER. *Iliad.* B.C. 1000

ALL factories use power and heat. Some use a lot of power and relatively little heat. Some use comparatively little power and a great deal of heat. Others use a great deal of both.

However efficient a condensing steam power plant is, it cannot convert much more than about *one-third* of the heat in its fuel into electrical energy. This has been explained in Section 88. More than half of the heat in the steam has to be wasted in the condenser. But quite a small plant can convert three-quarters of its fuel heat into steam heat, and all this heat can go to its process or to space heating.

If steam, on its way to the heating process, is first passed through an engine or turbine, a large factory can generate power with two-fifths of the coal used by a first-rate condensing power plant. Quite a modest engine in a small factory need only use two-thirds of the coal burnt by a condensing plant.

Many factories, using much power but little heat, generate their own power as a matter of course or custom. Unless these factories are using the exhaust steam from their engines or turbines, at least in part, they may be using two or three times as much coal per unit of power as the big power stations.

108. MODERN CONDENSING POWER PLANT EFFICIENCY. Fig. 29 shows the heat distribution in a medium pressure power station. (Two types of diagram are used—the " Sankey " diagram *a* and the " block " diagram *b*.) Of 100 parts of fuel heat only 29 parts are sent out as power. The rest is lost, chiefly as heat in the turbine exhaust for which there normally is no economic use.

Fig. 24 in Chapter 2 shows a heat cycle such as is used in many large modern power stations and which approximates to the Sankey diagram, Fig. 29.

It has a cycle efficiency of 44·3 per cent.

Let us assume that its turbines are 50,000 kW, so that their efficiency ratio may be 81 per cent.

The net turbine efficiency will be 81 per cent. of 44·3 per cent. = 35·9 per cent.

If the generator efficiency is 98 per cent., the turbo-generator efficiency will be 98 per cent. of 35·9 per cent. = 35·2 per cent.

A good boiler house will have an average efficiency of 85 per cent.

So that the overall coal to kilowatts efficiency will be 85 per cent. of 35·2 per cent. = 29·9 per cent.

If 4 per cent. of the current generated is used to drive the auxiliaries, lighting, fans, pumps, etc., the net overall efficiency will be

96 per cent. of 29·9 per cent. = 28·7 per cent.

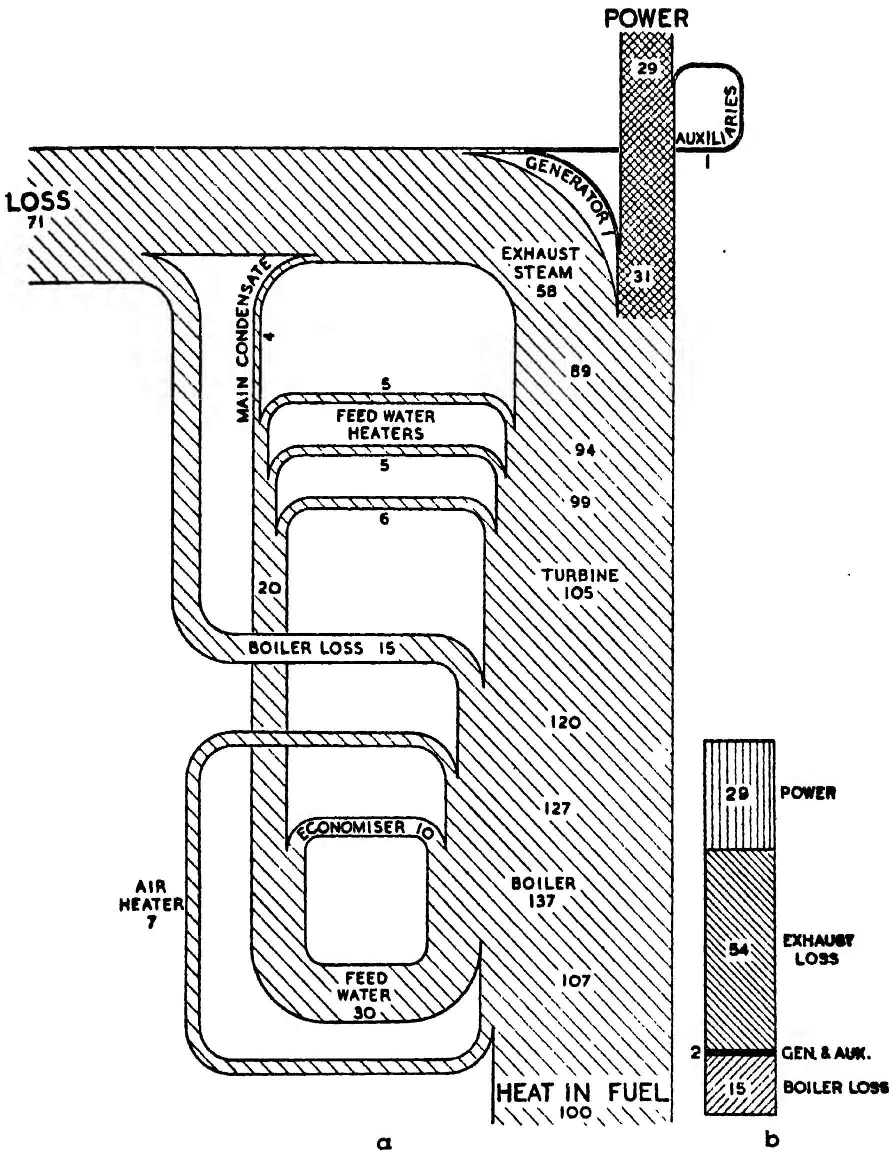


FIG. 29. ENERGY DISTRIBUTION IN A MEDIUM PRESSURE CONDENSING POWER STATION

(87664)

Let us take coal as containing 12,000 Btu per lb.

One kWh is equivalent to 3,415 Btu (Table VIII, Section 53).

In order to generate 1 kWh at 28·7 per cent. efficiency we will need to burn coal equivalent to $\frac{3{,}415}{\cdot 287} = 11{,}890$ Btu.

Or just under 1 lb. of coal per unit (actually ·99 lb./kWh).

Now this is good performance obtained from a first-rate station running continuously on base load.

There are a few stations working with higher efficiencies. The problem is one of finding a more efficient cycle that is workable. With steam reheating and six-stage regenerative bleeding it is possible to get a cycle efficiency of just over 50 per cent. The progress made is remarkable. Fifty years ago the cycle efficiency was about 15 per cent. This has been trebled, but it is still depressingly low.

Fig. 29a shows that 105 parts of heat go into the turbine although only 100 are contained in the fuel. This sometimes puzzles people. The answer is that the heat in the condensate and in the bled steam circulates round and round, and by so doing, increases the amount of steam that passes through the turbine, and thus generates more power.

In the same way the heat put in the economiser and air heater circulates round the boiler. It is heat that goes back into the boiler and is added to the heat of the fuel.

The big power station is only about 29 per cent. efficient. Now we all know that the big power station is very efficient. How can "29 per cent." and "very" be reconciled? The truth is that the big power station very efficiently operates a fundamentally inefficient process, because we have not yet been able to devise a generally useful process that can use the low temperature exhaust heat. During the war a process was found that could use power station exhaust. The power station to which this process was attached at once achieved an efficiency of 83 per cent. So that a big power plant may have an efficiency of 29 per cent. or 83 per cent. depending on whether the exhaust heat can be used or must be thrown away.

109. AVERAGE GRID EFFICIENCY. The big efficient stations rely in part for their efficiency on continuous running at almost constant load. To meet the peaks, stations have to be started up and run for only half the day or less. Naturally the more efficient stations are given the larger share of the running.

An old part-time station will probably have small units. The steam pressure may be about 250 psi and the superheat will be low. There will be no regenerative cycle so that the cycle efficiency is unlikely to exceed 30 per cent.

Take the efficiency ratio at 75 per cent.

Generator efficiency at 95 per cent.

Auxiliaries at 6 per cent. and boiler efficiency at 73 per cent.

The overall efficiency will be $30 \times \cdot 75 \times \cdot 95 \times \cdot 94 \times \cdot 73 = 14 \cdot 66$ per cent.

The coal consumption will be $\dfrac{3{,}415}{12{,}000 \times \cdot 1466}$ $1 \cdot 95$ lb. coal per kWh.

In both good and indifferent stations there is a further considerable loss for distribution before the current reaches the consumer.

The figures for 1956-7 in the Central Electricity Authority Report show that the average coal consumption per unit "sent out" was 1·28 and the average coal consumption per unit "paid for" was 1·4 lb./kWh. For the purpose of discussion in this Chapter the figure of 1·4 has been taken.

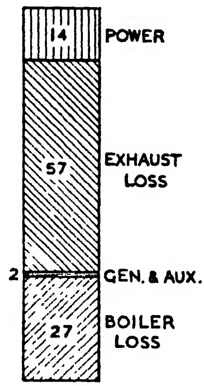


FIG. 30. PART-TIME POWER PLANT

A station running intermittently must always be less efficient than one running continuously. There will be starting and stopping losses, banking losses, the auxiliaries must do relatively more—some pumping and a lot of lighting are required even when the station is shut down. The result is that some of the peak current requires about $2\frac{3}{4}$ lb. of coal per unit, and the average coal consumption for the whole grid is nearly $1\frac{1}{2}$ lb. of coal per unit.

Fig. 30 shows a block diagram for a part-time station.

Some stations are now being specially built for two-shift working. One of these is showing results that compare very favourably with the base load stations.

110. HEATING EFFICIENCY. Heating by steam, if properly done with a suitable process, is very efficient. The only losses should be the boiler losses, distribution-radiation, and a minimum of leaks. We are not concerned here with the way in which the process steam is used—that will be dealt with in subsequent chapters; only with the supply of steam to the process.

The plant pictured in Fig. 31 has a large modern boiler plant with an efficiency of 83 per cent. The auxiliaries, fans, feed pump, etc., are taken as requiring 1 per cent. of the fuel energy. This is almost as much as that taken by a big power station and is manifestly wrong as there is no condenser water pumping and the feed pump will take much less power at lower boiler pressure; but the figure 1 per cent. was taken as it was the lowest round number for inclusion in Fig. 31. The condensate is all returned to the boilers after the flash heat—Section 44—has been used in the process. The overall efficiency of the heat delivered to the process is 82 per cent.

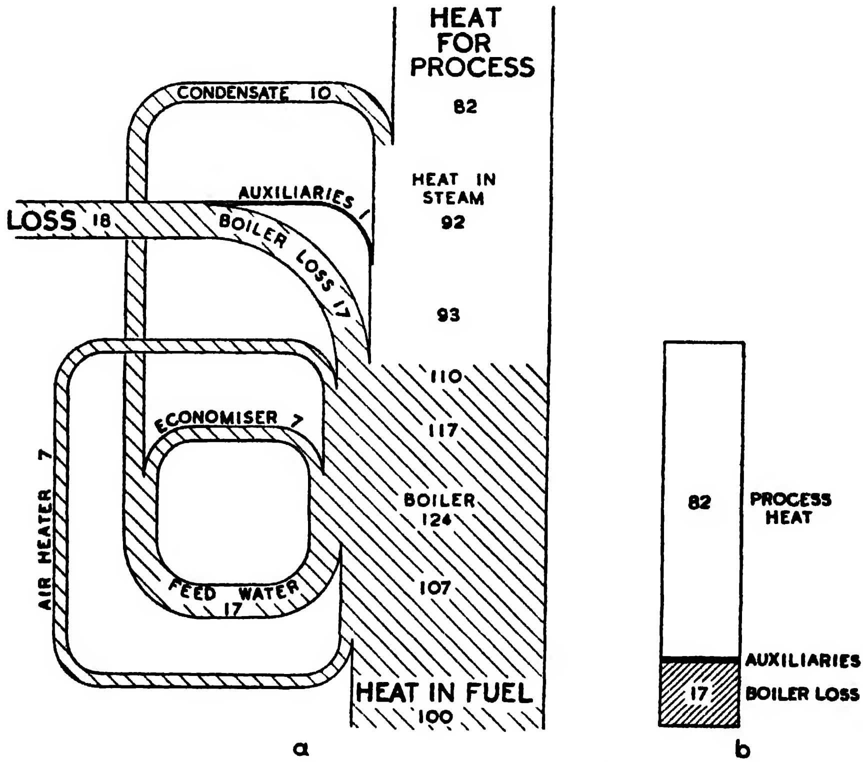


FIG. 31. LARGE EFFICIENT PROCESS STEAM PLANT

Fig. 32 shows a small inefficient heating plant. The boilers have only an efficiency of 50 per cent., 4 per cent. of the total steam is used and exhausted to atmosphere by the boiler feed pump. The condensate is all wasted. Yet the efficiency (heat to process/fuel heat) is 42 per cent.

111. COMBINED POWER AND HEATING. If steam on its way to process or space heating is passed through an engine or turbine a great economy in power generation is obtainable. Fig. 29 shows that three-quarters of the loss in a big power station is the heat in the exhaust steam. If this exhaust can be used for process heating it can almost be said that it does not matter how much

heat it contains (that is, how big the exhaust loss is) because it will not be lost. This is true up to a point only. The steam could have generated more power had we used it better.

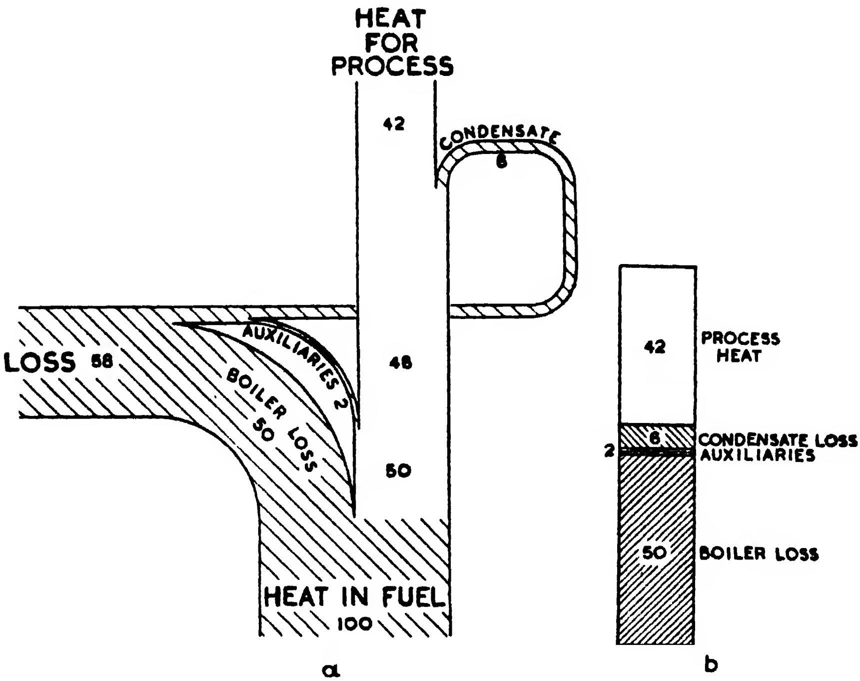


FIG. 32. SMALL INEFFICIENT PROCESS STEAM PLANT

112. POWER GENERATION LOSSES. The losses in an engine or turbine are made up of the following :—

1. Friction.
2. Radiation.
3. External leakage.
4. Piston leakage } (in reciprocating engines).
5. Valve leakage }
6. Interstage leakage } (in turbines).
7. Tip leakage }
8. Blade friction }
9. Gland leakage }
10. Wire-drawing (in both).

If the steam is being used for the production of power only, all these losses are real losses. But if the steam is all going to process and is merely generating power on its way then only the first three are definite irrecoverable losses, though No. 9 is often allowed to be a loss. Losses 4, 5, 6, 7, 8, 9 and 10 are only losses to power generation. No heat is lost by these losses. All the heat is still

there for use in the process. So, with certain limitations to be discussed later, we can say that apart from friction, radiation and external leakage IT DOES NOT MATTER THERMALLY HOW INEFFICIENT AN ENGINE IS PROVIDED THERE IS A REAL USE FOR ALL THE EXHAUST STEAM. Of course it does matter in practice. In most factories it is difficult or impossible to generate sufficient power. The more power that can be extracted from the steam before exhausting it the better. But the amount of power generated has practically no effect on the overall thermal efficiency provided *all* the exhaust steam is *usefully* employed.

Another important argument in favour of extracting all possible power from the steam is that for a constant power requirement this will reduce the steam passed to the process. There will thus be an urge on the process side to economise steam.

The efficiency of a cycle is shown by an available energy area on the temperature entropy diagram—see Chapter 2. The " inefficiency " is the area of the heat rejected to the exhaust. In a cycle where the exhaust heat is all used for process or space heating there is no rejected heat, so that the cycle or ideal efficiency is 100 per cent.

113. GOOD COMBINED HEATING AND POWER PLANT. In a large plant in which the process steam, instead of going straight to process, as in Fig. 31, passes first through a turbine, Fig. 33, the feed pump will use more power because the boiler pressure is higher. As, however, the energy for the auxiliaries in Fig. 31 is excessive, see Section 110, it will still be fair to take 1 per cent. of the heat input for the auxiliaries.

If we assume that the plant conditions are such (high boiler pressure, moderate back pressure) as to give 1 kWh for 26 lb. of steam,

and that the steam at the turbine inlet contains 1,300 Btu/lb.

The fraction of heat converted into power will be $\dfrac{3{,}415}{1{,}300 \times 26} = \cdot 1.$

We can therefore assume that we should convert 10 per cent of the heat input in our example into turbine power.

If 7 per cent. of this energy is lost in the turbine by radiation and friction, and if the generator is 97 per cent. efficient,

the actual power generated will be $10 \times \cdot 93 \times \cdot 97 = 9.$

One of these units is used for the auxiliaries, so that the net power output will be 8.

We can now build up Fig. 33a. This shows that we can pass the same heat to process as in Fig. 31, and can convert 8 energy units into power units for the additional use of 11 fuel energy units.

The effective generating efficiency is $\dfrac{8 \times 100}{11} = 73$ per cent.

This is $\dfrac{73}{29} = 2 \cdot 5$ times as good as the big condensing station.

The coal consumption will be $\dfrac{3{,}415}{12{,}000 \times \cdot 73} = \cdot 39$ lb. coal/kWh.

The effective generating efficiency is 73 per cent.,

but the overall thermal efficiency is $\frac{(82 + 8)\ 100}{111} = 81$ per cent.

The reason the overall efficiency is higher than the generating efficiency is because the turbo-generator losses and the auxiliary losses have been debited to power generation as well as to overall heat use and these losses represent a much higher proportion of the power energy.

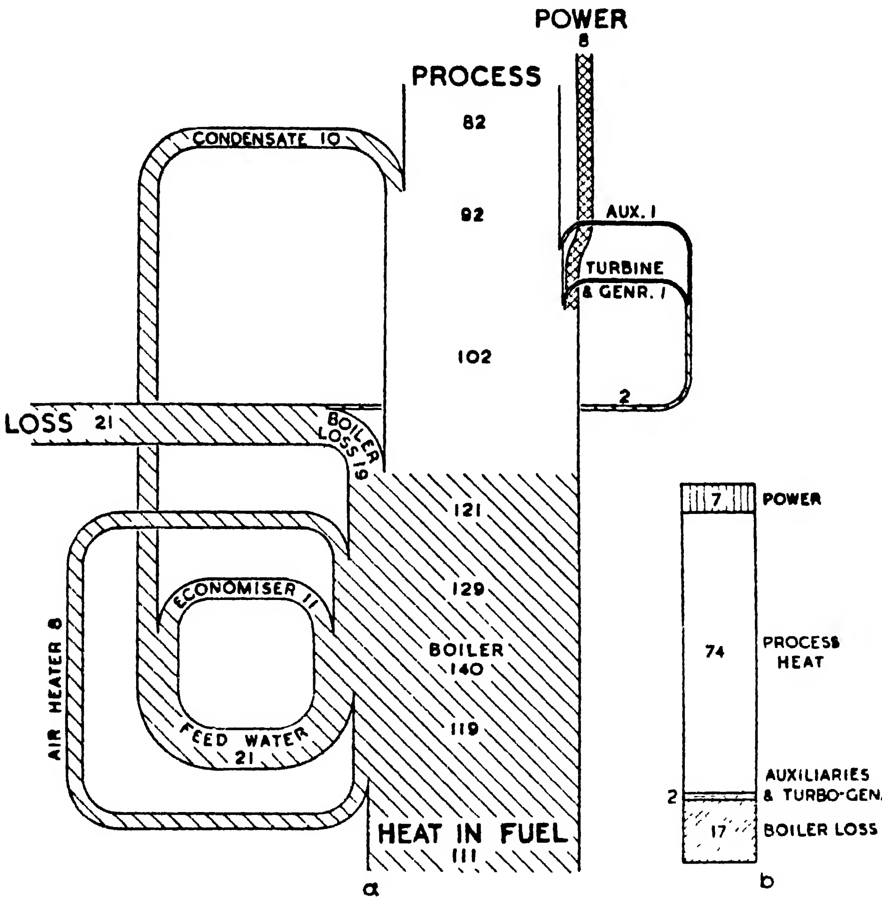


FIG. 33. LARGE EFFICIENT COMBINED POWER AND HEATING PLANT

It may be noticed that the economiser figures in Figs. 31a and 33a do not tally. This is because with the low boiler pressure in Fig. 31 there is less scope for an economiser and it is assumed that with the lower boiler temperature the generating surface can be arranged to compensate for the smaller economiser.

In Fig. 33 lower efficiencies have been taken all over than in Fig. 29. In particular, the whole of the auxiliaries have been debited against power

generation, when the only fair debit is the extra power needed by the boiler feed pump at the higher pressure. Large industrial plants can and do operate at the same efficiencies as the power stations. Liberal figures have been taken in order to be conservative.

Fig. 33b shows the affairs of Fig. 33a in block diagram percentage form.

Now 8 power units obtained from 11 fuel heat units is equivalent to burning ·39 lb. of coal per electrical unit. This must be compared with just under 1 lb. of coal for the big power station and 1·4 lb. of coal for the grid average.

114. INEFFICIENT COMBINED POWER AND HEATING PLANT. In the small plant we will assume that the steam consumption would be 50 lb./kWh, and that the inlet steam contains 1,200 Btu/lb.

We are therefore getting 3,415 Btu in electricity from 1,200 × 50 Btu of steam heat.

As, in Fig. 32, we have assumed a boiler efficiency of 50 per cent. we shall get 3,415 Btu of electricity from 1,200 × 100 Btu of coal heat.

We can therefore expect to convert $\frac{3{,}415 \times 100}{1{,}200 \times 100}$ or about 3 heat units into power units.

We shall leave the steam-driven feed pump alone.

Let us assume that due to leakage, radiation and generator inefficiency we lost 1 heat unit—that is 33 per cent. of the power available—then we shall get a net power output of 2 heat units.

We can now build up Fig. 34a. We see that we get the same power output as the energy taken by the deplorable feed pump. We also see that by burning 6 extra fuel heat units we can convert 2 into power.

The generating efficiency is $\frac{2 \times 100}{6} = 33$ per cent.,—rather better than quite a big condensing station.

The coal equivalent is $\frac{3{,}415}{12{,}000 \times \cdot 33} = \cdot 86$ lb. coal/kWh.

The overall thermal efficiency of this plant is $\frac{(42 + 2)\ 100}{106} = 41$ per cent.

The efficiency of the bad small combined plant equals or exceeds power station efficiency because the process that combines power generation with heating has 100 per cent. theoretical overall efficiency. The highest theoretical efficiency of a straight condensing power station is about 50 per cent. So that the power station works under a great handicap.

It must, however, be remembered that the efficiency of the small bad combined plant appears to equal or exceed the efficiency obtainable from a condensing power station because it is assumed that 100 per cent. of the exhaust steam is usefully employed in a process. In only too many factories much of the process steam is wasted. Were the process extravagances

eliminated, it would perhaps be necessary to exhaust in part to atmosphere or to a condenser. In such circumstances the generating efficiency would drop greatly.

Table XI compares the figures side by side.

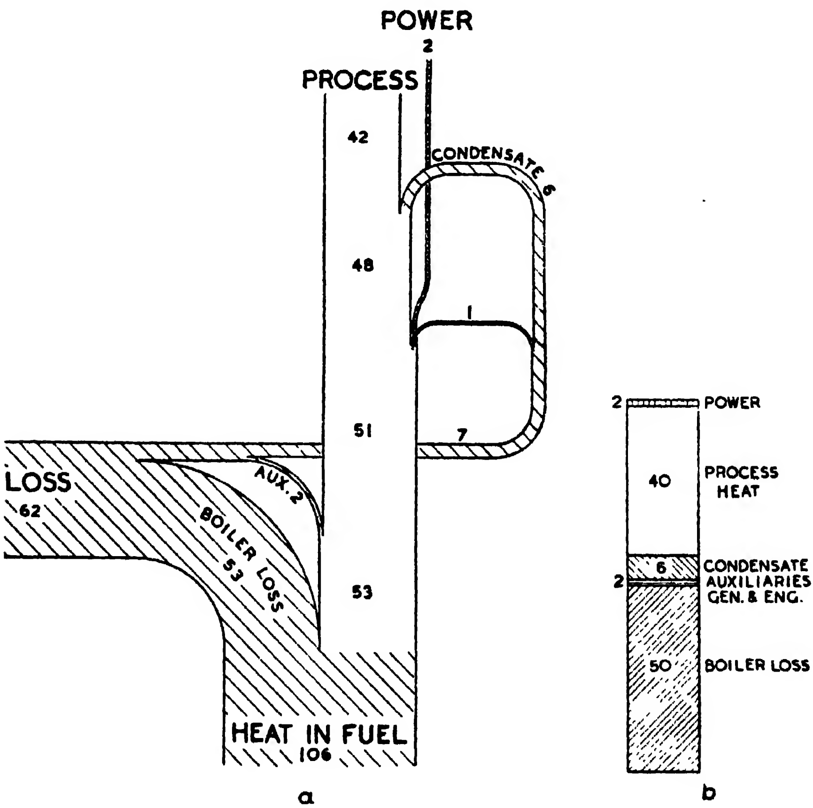


FIG. 34. SMALL INEFFICIENT COMBINED POWER AND HEATING PLANT

TABLE XI. COMPARATIVE POWER GENERATING EFFICIENCY

<table>
<tr><th></th><th>GOOD CONDENSING POWER PLANT</th><th>BAD SMALL COMBINED POWER AND HEATING PLANT</th><th>GOOD LARGE COMBINED POWER AND HEATING PLANT</th></tr>
<tr><td>Theoretical process efficiency</td><td>45%</td><td>100%</td><td>100%</td></tr>
<tr><td>Boiler efficiency .. ..</td><td>85%</td><td>50%</td><td>83%</td></tr>
<tr><td>Machine loss .. ..</td><td>5%</td><td>30%</td><td>10%</td></tr>
<tr><td>Coal consumption .. ..</td><td>·99 lb./kWh</td><td>·86 lb./kWh</td><td>·39 lb./kWh</td></tr>
<tr><td>Generating efficiency ..</td><td rowspan="2">28·7%</td><td>33%</td><td>73%</td></tr>
<tr><td>Overall efficiency .. ..</td><td>41%</td><td>81%</td></tr>
</table>

115. OTHER INDUSTRIAL STEAM ENGINES. Many factories, particularly old-established mills, generate their own power as a matter of course or custom. If the exhaust steam can be used in whole or in part, this may be economical. If however the exhaust heat is thrown away into a condenser or into the atmosphere, it is certainly against the National interest although it may pay. Let us look for a moment at two cases. Take a 1,000-H.P. engine exhausting into a condenser at 20-in. vacuum and taking saturated steam at 200 psi.g.

As the engine is taking saturated steam, the exhaust must be wet, so that Gibbs' Function can be used in the working out.

The total heat in 200 psi.g. steam is 1,200 Btu/lb. with an entropy of $1 \cdot 541$.

The temperature of 20-in. vacuum steam is 161° F. and Gibbs' Function is 16.

The total heat at exhaust after adiabatic expansion will be

$$(460 + 161) \times 1 \cdot 541 - 16 = 941 \text{ Btu.}$$

The adiabatic heat drop is $1{,}200 - 941 = 259$.

The heat input is 1,200 less the heat in the feed water—say 125 Btu.

The cycle efficiency will be $\dfrac{259 \times 100}{1{,}200 - 125} = 24 \cdot 1$ per cent.

From the information given in Section 104 we can estimate the efficiency ratio of a 1,000-H.P. engine operating between 200 psi.g and 20-in. vacuum as 55 per cent.

Let us take the boiler efficiency at 70 per cent., the auxiliaries at 3 per cent., then the overall efficiency will be $24 \cdot 1 \times \cdot 55 \times \cdot 70 \times \cdot 97 = 9 \cdot 0$ per cent.

The coal consumption will be $\dfrac{3{,}415}{12{,}000 \times \cdot 09} = 3 \cdot 16$ lb./kWh.

We can construct Fig. 35 from this data.

Now the average of all the grid stations is about $1 \cdot 4$ lb. coal per electrical unit at the customer's premises. In order to compare grid current with the engine we must allow for the inefficiency of the motors that will replace the engine. Taking the electric motors at an average of 85 per cent. efficiency the grid current becomes, at the factory line shaft, $1 \cdot 65$ lb. coal/kWh. It is clear that from the National point of view in this factory, factory generation is inefficient. If the engine is running principally at grid peaks then it may not be nationally uneconomic to use the engine ; but if the engine is running 24 hours a day then it is most certainly against the National interest and is probably against the factory's economic interest to use the engine.

Very large industrial condensing turbine plants are sometimes more efficient than the power stations, because the plant in the big factory is just the same as in the power station and the distribution loss is saved. In some factories the load factor is higher than that of many utility undertakings.

116. DIRECT ACTING PUMPS. Nearly every steam using factory has direct acting steam pumps. These pumps have an efficient pumping end, but a very inefficient steam end. Even if there are no process pumps, the boiler

feed pump in most small factories is a direct acting pump exhausting, only too often, to atmosphere.

A direct acting pump works on a non-expansive cycle—see Sections 69 and 78—and consequently uses none of the internal energy of the steam. The work available is simply the product of the pressure and the increase in volume of the water during evaporation.

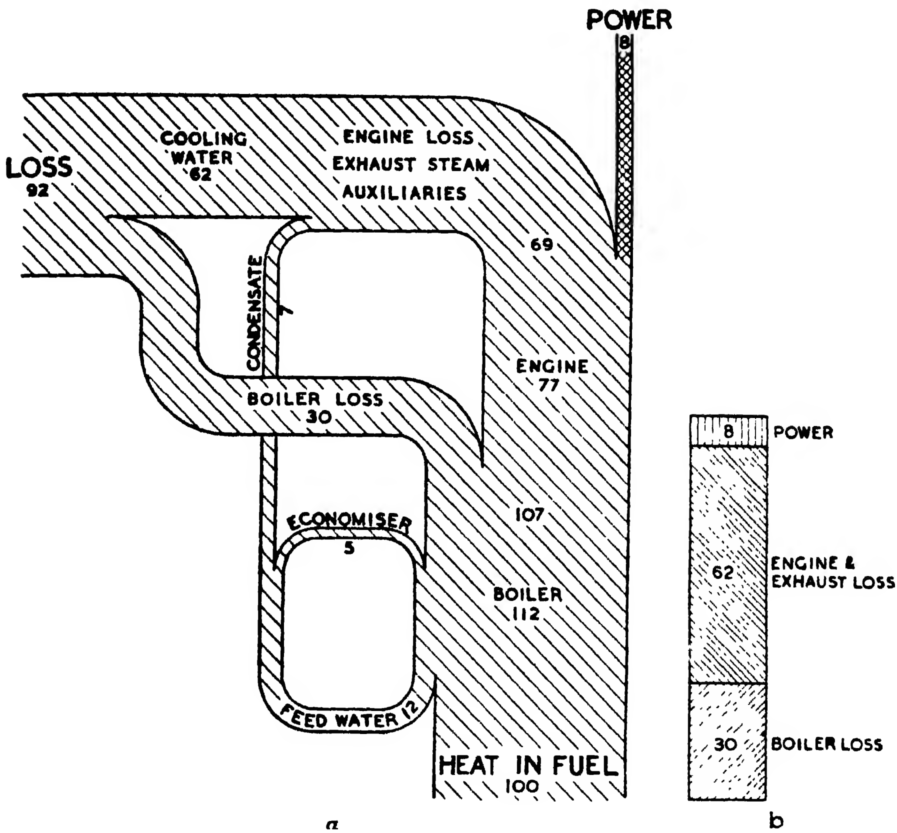


FIG. 35. 1,000-H.P. INDUSTRIAL CONDENSING ENGINE

The increase of volume of 1 lb. of water at 150 psi.g. to 1 lb. of steam is 2·74 cu. ft.

The external work done per lb. of steam will be

$$150 \times 2{\cdot}74 \times 144 = 59{,}184 \text{ ft. lb.,}$$

or, dividing by 778 (see Table VII), 76 Btu.

The total heat of saturated steam at 150 psi is 1,197 Btu/lb., so that the cycle efficiency is $\frac{76 \times 100}{1{,}197}$ 6·35 per cent.

If the efficiency ratio of the steam end of the pump is 50 per cent. and the boiler efficiency is 65 per cent. the net efficiency will be 2 per cent.

This deplorable figure is equivalent to $\frac{3{,}415}{12{,}000 \times .02} = 14$ lb. coal/kWh.

The heat distribution of the steam end only of this terrible machine is shown in Fig. 36.

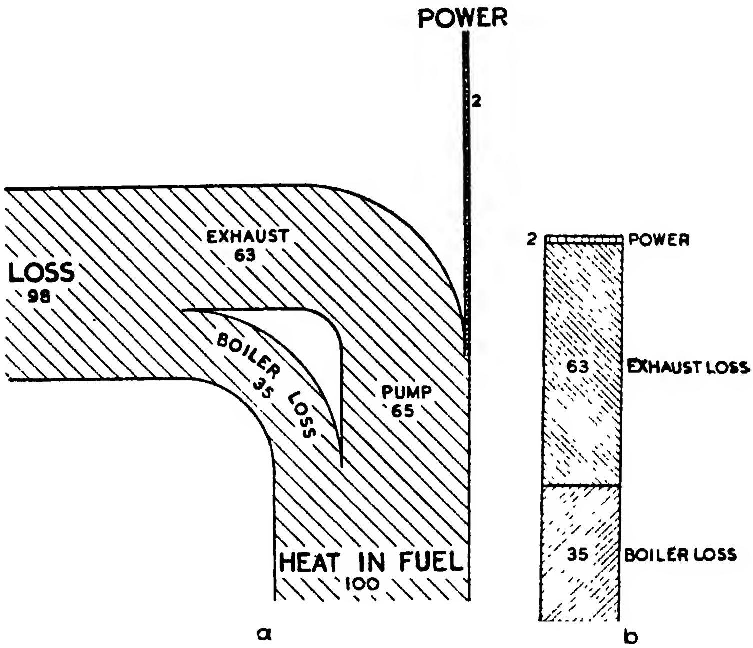


FIG. 36. NON-EXPANSIVE NON-CONDENSING PUMP

Some direct acting pumps have a valve gear that allows for a slight amount of expansive working, but in practice the bye-pass steam has usually to be used to make the pump complete its stroke.

If the pump is replaced by an electrically-driven pump a great improvement in efficiency is secured at the driving end, but things deteriorate at the pumping end. If a reciprocating pump is employed the pump discharge must always be in excess of maximum demand, and when reduced quantities are needed the surplus must be bye-passed. This is wasteful. If the reciprocator is replaced by a centrifugal pump a serious drop in pump efficiency occurs, especially when pumping against high heads. The churning against a checked discharge valve wastes much power. The easy way out is to retain the steam pump and to use the pump exhaust for heating.

117. THE EVILS OF REDUCING VALVES. The process carried out by a reducing valve is shown in Fig. 17 and is described in Section 74. All that a reducing valve does is to increase entropy without getting anything in exchange.

Reducing valves are, of course, often essential, but their use should be reduced to the absolute minimum.

A reducing valve is, from a thermodynamic point of view, an invention of the devil. It sets out to degrade good heat ; to dissipate the good high potential. It performs this vile task to perfection until it goes wrong. The use of a reducing valve might be called an admission of defeat. It is the easy way out. A reducing valve can often be replaced by one of three things : an engine, a steam accumulator or an evaporator. The substitution of engines for reducing valves is the main theme of this Chapter. Where the pressure drop or the steam quantity does not warrant the use of an engine, an accumulator may level out peaks and prevent safety valve blows and pressure drops.

Many factories would improve their processes very much had they an ample supply of distilled water. Some factories have to throw away much of their condensate because it gets contaminated in the process—for example, in the curing of rubber. An evaporator in place of a reducing valve would supply the missing condensate as distilled water. The evaporator will supply the required low pressure steam to process while the high pressure steam is turned into pure condensate.

This process increases entropy and degrades the good potential in the high pressure steam ; but it gives us something useful in return. Accumulators and evaporators are dealt with in subsequent chapters.

A reducing valve is sometimes excused on the grounds that the wire-drawing dries wet steam. This is true, but the drying effect is very small and the process is overrated, *see* Section 182.

The distinction between a surplus and a reducing valve needs clarification because many of the so-called reducing valves in manufacturers' catalogues are not reducing valves but are surplus valves. A true reducing valve is a valve which maintains a steady pressure on the low pressure side of the valve and whose sole impulse is obtained from the low pressure. A surplus valve is simply, in principle, a safety valve discharging into a low pressure pipe. When the pressure on the high pressure side of the valve reaches a certain value the valve opens. Many valves, such as the Ruths accumulator valve combine surplus and reducing qualities. It is no use buying a valve that is called a reducing valve when it is really a surplus valve, and expecting it to provide a steady pressure in the low-pressure main.

Ways of getting rid of reducing valves by means of special engines or turbines, called " pass-out " machines, are described in later sections.

Reducing valves, like any other plant, can occasionally go wrong and stick open. It is therefore essential that the plant fed by the reducing valve be provided with an adequate safety valve. This precaution must NEVER be forgotten.

118. COMPARATIVE GENERATING EFFICIENCIES. It is interesting and very instructive to tabulate the coal consumed per electrical unit with various methods and machines. Table XII assumes that the whole of the exhaust steam from the back pressure machines is being usefully employed in the process.

(An engine that generates power and exhausts to process must work against a higher exhaust or " back " pressure than one exhausting to atmosphere or into a condenser. An engine or turbine exhausting to process is called a " Back Pressure " machine.)

119. BOUGHT POWER. Table XII shows that from a National point of view to safeguard our dwindling coal assets, every steam-using factory should, if possible, raise steam at such a pressure that the steam can first generate power before being passed to the process.

Whether it pays from a £ s.d. point of view to buy power or to generate it depends solely on the cost of power generation and on the price that must be paid for the public supply.

TABLE XII. APPROXIMATE COAL CONSUMPTION OF POWER GENERATION

	APPROX. LB. COAL PER KWH
Back pressure—large good	·4
Back pressure—small bad	·9
Public supply—base load	1·0
Public supply—average	1·4
Public supply—peaks	2·7
Large industrial condensing engine ..	3·2
Non-expansive non-condensing pump ..	14·0

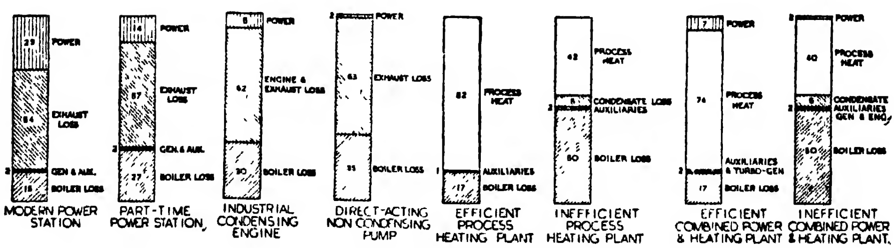


FIG. 37. COMPARATIVE METHODS OF POWER GENERATION

The cost of back pressure power is dealt with in the next section. In comparing these costs with bought power the purchased current must be brought to a comparable basis. That is to say, maximum demand charges, whether imposed as an inducement to use more current or for discouraging the overloading of a distribution or generating system, must be converted to cost per unit. If the tariff is also complicated with power factor and coal clauses the real cost can only be obtained by dividing the total money paid to the Supply Company by the units used.

The block diagrams from Fig. 29–36 are collected together for easy comparison in Fig. 37.

120. THE COST OF BACK PRESSURE POWER. We have seen, in Sections 113 and 114, that the coal used in the production of electrical power by means of back pressure plant will probably lie between ·4 and ·8 lb./kWh. Coal at 100*s*. a ton is worth ·536*d*./lb. So that back pressure electricity will cost between ·2 and ·4*d*. per unit for coal only. To this must be added the charges on the capital needed to pay for the extra plant including any necessary provision for standby or week-end supply. There are no extra running charges of any size on the boiler plant. The boilers and their staff have to be there anyhow. Maintenance and engine driver must be included.

Table XIII shows the costs in two London Factories, the author's and a friend's.

TABLE XIII. BACK PRESSURE POWER COSTS

	Winter 1943–44		Winter 1957–58
Size of plant—kW	350	4,000	9,000
Hours per week	70	140	130
Coal consumption—lb./kWh	·6	·4	·4
Coal cost	·15	·10	·24
Overheads and depreciation	·26	·09	·12
Maintenance	·08	·03	·04
Labour	·12	·03	·04
Miscellaneous	·02	0·2	·01
	·63 d./kWh	·27 d./kWh	·45 d./kWh

In the case of the smaller set the factory is only in steam/power balance in the winter. During the summer the steam load is much reduced, some current has to be taken from the grid and the generating cost goes up. In the case of the author's factory there is little difference between summer and winter conditions.

We have seen in previous Sections that when steam passes through a back pressure engine or turbine it suffers a heat drop. This heat drop represents the energy turned into useful power together with the energy needed to make good the following losses :—

Leakage.
Radiation.
Friction.
Generator loss.

Any leakage is a dead loss to process. How far it is a loss to power depends on the point in the machine where the leak occurs. In any case leakage loss must be debited to power generation because the leak is caused by the attempt to generate power.

The adiabatic heat drop, as calculated or found on a Mollier diagram, when divided into 3,415 (the electrical equivalent of heat) gives the ideal steam consumption of a perfect machine. Suppose the actual steam consumption is double the ideal; this means that the machine is losing half the available heat drop. The loss of heat drop is only lost to power generation, most of it goes out in the exhaust to process.

It is difficult to calculate in advance the real heat used by a back pressure machine. The adiabatic heat drop has nothing to do with it at all—though some engineers try to calculate back pressure power cost from the adiabatic heat drop and get very wrong results in consequence. The difference between the adiabatic heat drop and the actual heat drop is energy lost to *power generation*, not that lost to the power-cum-process as a whole. Most of the adiabatic loss goes out in the exhaust steam which goes to process and is not lost.

Some engineers take the view that by taking the ideal steam consumption and the actual steam consumption they can find the heat drop—this is not true in a back pressure machine. An absurd example will make this clear. Suppose an engine is generating power with 30 lb. steam/kWh. Now suppose the steam consumption is raised to 40 lb./kWh in two ways, first by drilling a small hole in the piston and second by removing the cylinder lagging and spraying cold water on the cylinder. In both cases 10 lb./kWh is lost to power generation, but in the first case no heat is lost, it goes through the hole to process; in the second case the heat consumption, and therefore the steam cost, is increased four-fold because 10 lb. of steam per kWh is condensed and its heat is lost to the process. No Mollier diagram or steam meter can distinguish between these processes in the absence of measurement of the exhaust steam quality.

Therefore to ascertain back pressure coal consumption it is no use taking notice of the steam consumption unless the exhaust quality can be measured. The only heat consumed is the electrical heat equivalent and the true machine losses.

If there is no appreciable leakage the real heat drop over the machine can be found by taking a sample of the exhaust steam and measuring its quality by calorimeter. The measurement of steam quality, or "calorimetry", is dealt with in Sections 213 and 234. The coal consumption per kWh is

$$\frac{\text{Steam flow} \times \text{Real heat drop} \times 100}{\text{Electrical output} \times \text{C.V. of coal} \times \text{Boiler efficiency}}$$

In the 4,000 kW machine referred to in Table XIII the exhaust is superheated and the total heat in the exhaust steam is easily found by means of a thermometer and a pressure gauge. (If the exhaust is wet, both sampling and calorimetry present great difficulty.)

The following figures were obtained on a light load :—

Steam flow (corrected for H.P. gland) ..	71,500 lb./hour.
Initial total heat	1,325 Btu/lb.
Exhaust total heat	1,202 Btu/lb.
Electrical output	2,250 kW.
Calorific Value of coal..	12,000 Btu/lb.
Boiler efficiency..	81 per cent.

This gives :—

$$\frac{71{,}500 \times (1{,}325 - 1{,}202) \times 100}{2{,}250 \times 12{,}000 \times 81} = \cdot 402 \text{ lb. coal/kWh.}$$

Now a correction has been made in the foregoing calculation for the steam that leaks from the damaged H.P. gland and is piped to the 10 psi main. The actual steam consumption was 75,000 lb. and the metered H.P. gland leak was 4,000 lb./hr. Some of the gland leak steam will have passed through the first wheel and done some work. It has been assumed that the work done on the first wheel by the gland leak steam is equivalent to 500 lb./hr. passing through the whole turbine, so that the steam consumption has been taken as being $75{,}000 - 4{,}000 + 500 = 71{,}500$ lb./hr.

On the other hand some small amount of steam actually does escape into the engine room and this steam has been assumed to have gone to process. But the quantity is so small that it can safely be ignored.

Had the gland steam gone to atmosphere or to a condenser its loss should have been debited in toto. Assume that 4,000 lb. steam/hour had blown to atmosphere from the H.P. gland, then $4{,}000 \times 1{,}325 = 5{,}300{,}000$ Btu must be added to the heat used in the above fraction. (This assumes that the leaking steam has done no work in the first wheel.) The fraction then becomes :—

$$\frac{\left\{5{,}300{,}000 + (71{,}500 \times 123)\right\} 100}{2{,}250 \times 12{,}000 \times 81} = \cdot 644 \text{ lb. coal/kWh.}$$

This shows how very important it is to reduce leaks to a minimum, and, where leaks may be large, to catch the leak and use it. If the gland leak had been ignored the calculation would have been

$$\frac{75{,}000 \times 123 \times 100}{2{,}250 \times 12{,}000 \times 81} = \cdot 422 \text{ lb. coal/kWh.}$$

If there is no way of measuring leakage the losses must be estimated by an experienced engineer or the makers should be asked to give an estimate ; but the efficiency ratio is useless, only the real losses are wanted, namely :—

Radiation.	True leaks.
Friction and windage.	Generator losses.

TABLE XIV. APPROXIMATE BACK PRESSURE POWER HEAT CONSUMPTION

SIZE OF SET KW	BTU TAKEN FROM STEAM PER KWH
3,000 and over	$3{,}415 \times 1\cdot 1$
1,000 to 3,000	$3{,}415 \times 1\cdot 125$
250 to 1,000	$3{,}415 \times 1\cdot 15$
100 to 250	$3{,}415 \times 1\cdot 175$
50 to 100	$3{,}415 \times 1\cdot 2$
Less than 50	$3{,}415 \times 1\cdot 25$

In the absence of any real information, Table XIV gives a rough indication of what may be expected from machines in first-class order, where the gland steam of turbines is used for low pressure process purposes.

121. THE WILLANS LINE. There is an important point about the steam consumption of engines that must be considered before we investigate further the best way of generating power.

Fig. 38 shows the steam consumption of a 350 H.P. steam engine working with an initial pressure of 200 psi.g. against a back pressure of 20 psi.g. From the curve we can tabulate the steam consumption thus :—

Horse Power	*Steam lb./hr.*	*Steam lb./H.P./hr.*	*No-Load Steam*	*Useful Steam*	*Useful Steam lb./H.P./hr.*
50	4,550	91	3,700	850	17
100	5,400	54	3,700	1,700	17
150	6,300	42	3,700	2,600	17
200	7,100	35	3,700	3,400	17
250	8,000	32	3,700	4,300	17
300	8,800	29	3,700	5,100	17
350	9,700	28	3,700	6,000	17

The very high no-load consumption of this particular engine is due to its working against a back pressure. At no load it is taking steam not only to make up for losses, but also to push the piston up and down against the back pressure. This question is discussed more fully in Chapter 7.

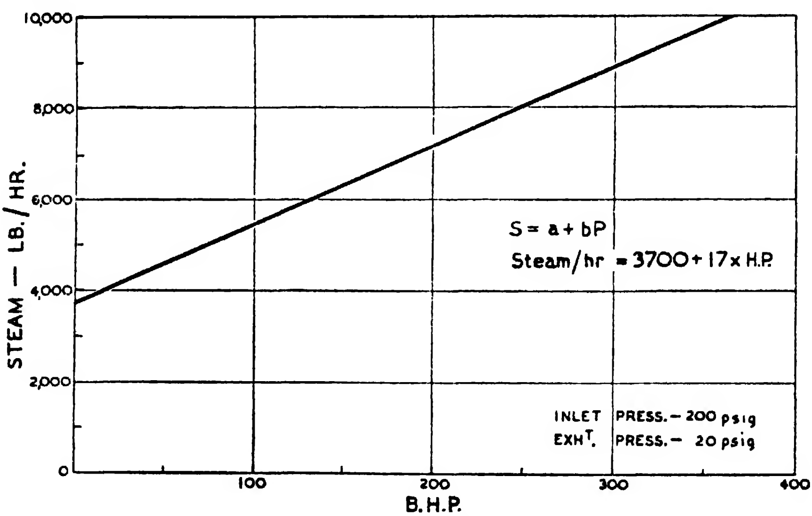


FIG. 38. THE WILLANS LINE

This tabulation shows that just to make up for the losses, keeping the engine warm, overcoming friction, pushing the piston up and down, etc., requires 3,700 lb. of steam per hour. Were the engine exhausting into a condenser

less steam would be taken. At all loads it is very interesting to see that the actual power producing steam is constant at 17 lb./horse power/hour. This constant steam consumption per horse power can be proved theoretically and is known as Willans Law, and the graph showing the steam consumption of an engine as a straight line with its origin at the no-load consumption is called the "Willans Line" (after P. W. Willans who discovered this relation in 1888).

The steam consumption of an engine or turbine is therefore made up of two parts. A constant quantity which turns the engine, overcomes the back pressure and makes up for the losses—this is the "fixed" steam; and a quantity which is directly proportional to the output of the engine and which actually does the work—this is called the "marginal" steam.

The Willans Line is only straight in a turbine or in an engine that is throttle governed. If the engine is cut-off governed the line is curved. Calling the steam consumption S and the horse power P the formula for the Willans Line is:—

$$S = a + bP$$

where a is the "fixed" steam and b is the variable or "marginal" consumption per H.P. In the particular engine under consideration the formula is:—

$$S = 3{,}700 + 17P$$

It is possible to make the Willans formula rather more useful by taking readings at different back pressures and by introducing a back pressure term, provided the back pressure is the principal cause of the fixed steam consumption, as it is in a turbine. In the case of the 4,000 kW turbine in the author's factory the Willans lines at various back pressures enable the following formula to be used with very fair accuracy over the ordinary running range:—

$$S = (360 \times \text{Absolute Back Pressure}) + (23{\cdot}5 \times \text{kW})$$

Suppose that the process requires 100,000 lb. steam per hour at a back pressure of 70 psi.g. the power that the turbine can give will be a maximum of 2,950 kW. If the back pressure can be reduced to 50 psi.g. an additional 310 kW can be generated from the same amount of steam.

122. WILLANS LINE FOR TWO ENGINES. Fig. 39 shows the steam consumption plotted against output for a plant with two small engines. Each set has an output of 175 kW and the no-load steam consumption is 3,000 lb. steam/hour. In this factory the normal process demand in summer is 6,000 lb./hr. This is well within the capacity of one set. In winter the steam demand rises to 12,000 lb./hr. and it is then necessary to run both sets, so as to use the power producing properties of the steam to the full. It will be seen that if more than 175 kW are needed when only one set is running, the starting up of the second set will cause a sudden extra output of 3,000 lb. steam/hr. from which no power whatever is produced. If much running has to be done with a set-and-a-quarter or a set-and-a-half in summer, it would be much more economical to have a larger engine which, although it would have a larger no-load consumption, would not have so large a no-load consumption as the two small sets combined, and its marginal consumption would be less. The broken line in Fig. 39 is the Willans Line for one 350 kW set.

123. THE STEAM/POWER RATIO. Industries fall roughly into three categories. The first, large steam users whose steam demand is so great that they can generate more than sufficient power for their needs. The second group comprises those factories whose power and steam requirements are in balance. The third, those industries whose power requirements are greater than can be met by using back pressure generating machines. There is a constant flow of factories from the first and second categories towards the second and third. The more efficient the factory the more certain and speedy is its

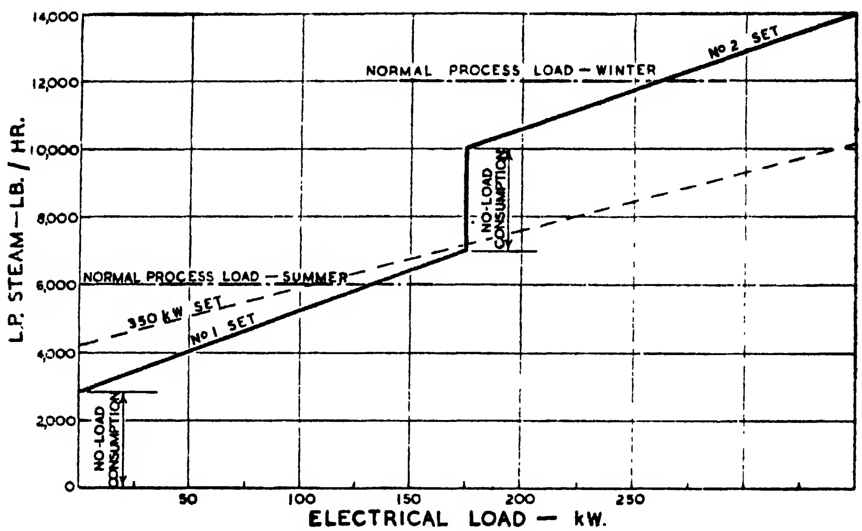


FIG. 39. WILLANS LINE FOR TWO ENGINES

transition from adequate power producing capacity to definite power shortage. The reasons for this drift are simple. Every economy of steam in the heat-using processes reduces the amount of steam available for power generation. Most technical improvements reduce steam demand and increase power demand. Every labour-saving improvement increases power demand.

Some industries have reached the point where attempts to balance steam and power are now hopeless in an efficient factory, and the fact is accepted, as a matter of course, that some power must be generated without using the exhaust steam—such an industry is the paper industry.

Some industries are only just beginning to realise that there is a steam-power problem. The average Laundry can generate all the power it needs from its process steam, but a really go-ahead Laundry has difficulty in supplying its power needs.

124. RECONCILIATION OF STEAM AND POWER DEMANDS. In that large group of factories where the demand for power exceeds the amount which can usefully be generated by means of back pressure working alone,

the difference must be made up by one or a combination of the following methods :—

(*a*) By lowering the back pressure.
(*b*) By blowing off exhaust steam.
(*c*) By installing a separate condensing set.
(*d*) By buying electricity from the public supply.
(*e*) By installing a pass out machine.
(*f*) By installing an exhaust turbine.
(*g*) By installing a mixed pressure turbine.
(*h*) By installing a vacuum turbine.
(*i*) By installing some form of heat accumulator.
(*j*) By increasing the feed water temperature.
(*k*) By supplying exhaust steam to a neighbour.
(*l*) By running a power economy campaign.

There may be some machines named in this list whose functions are not obvious. Here are definitions :—

Condensing Engine or Turbine. A machine exhausting into a condenser and taking steam at boiler pressure unless otherwise qualified.

Back Pressure Engine or Turbine. A machine taking steam at boiler pressure and exhausting into a pipe which leads neither to a condenser nor to atmosphere, but to another machine or process.

Pass-out, Bleeder or Extraction Engine or Turbine. A machine where, at some point intermediate between inlet and exhaust, some steam is extracted or passed out at a pressure above exhaust pressure.

Mixed Pressure Turbine. A machine supplied with more than one steam supply at, generally, widely differing pressures.

Exhaust Turbine. A turbine whose only steam supply has been exhausted from another turbine or engine.

Vacuum Turbine. A turbine taking steam from a process at below atmospheric pressure.

These machines are often combined and their functions will be clear from their names. For example, " A Multiple Pass-out Back Pressure Engine " means that the engine exhausts to a process or another machine, and that steam is drawn off from the engine at more than one pressure above exhaust pressure—this means that the engine must necessarily be triple or quadruple expansion. Or again, " A Pass-out Exhaust Condensing Turbine " means a machine whose steam supply is the exhaust from another machine, which exhausts into a condenser and which passes out a supply of steam at some pressure intermediate between admission and exhaust.

125. CHOICE OF BACK PRESSURE. The advantages for power production of raising the boiler pressure are fairly obvious, but a much greater gain can be secured by reducing the exhaust pressure. Almost exactly the same gain results from reducing the exhaust pressure from 15 psi.a. to 1 psi.a. as by raising the initial pressure from 100 to 1,000 psi. Fig. 40 shows the advantage of raising the initial pressure and lowering the back pressure by 10 psi on an engine working between 150 and 40 psi.a. This is a pressure/volume or ideal indicator diagram and the area shows the power available.

The reason the gain from raising the initial pressure is so small is that this increased pressure only operates over about quarter of the stroke, whereas the reduced back pressure operates during the whole stroke. In the same way the gain with increased initial pressure operates on the small end of a turbine while reduced back pressure operates on the big end.

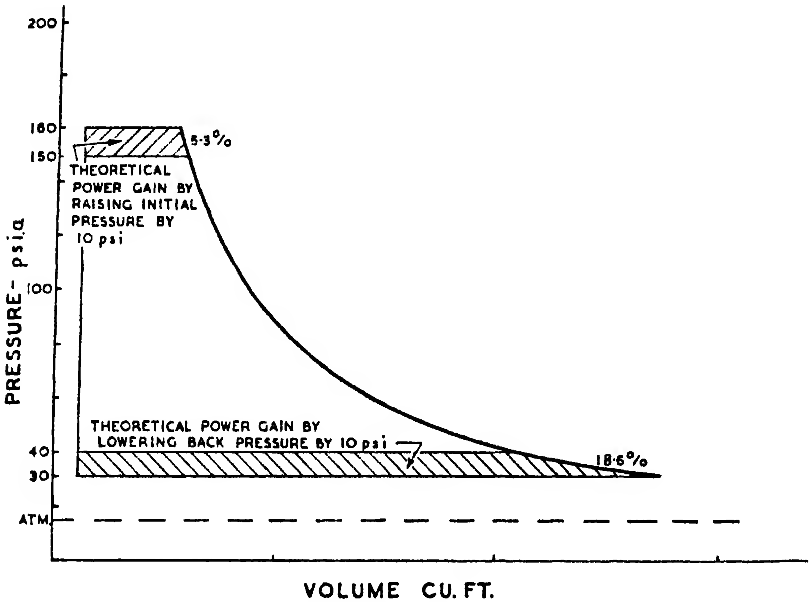


FIG. 40. COMPARATIVE GAINS BY ALTERING INITIAL AND EXHAUST PRESSURES

In considering the installation of back pressure plant the first thing to settle is the back pressure—the lower the better. It may be that steam is being raised at 150 psi, fed to a reducing valve where the pressure is reduced to perhaps 80 psi, and then distributed to various points where it is still further reduced.

Whatever the pressures actually in use, every effort should be made to reduce them. In many places in this book the virtue of using the lowest possible process pressure is proclaimed. If each process is separately examined, it is seldom that some reduction in pressure is not found to be possible. The use of steam for process is to put heat into a product. The hotter the product, the hotter (and therefore higher pressure) must the steam be. So the first thing is to ensure that the processing temperature is as cool as possible. Only a few processes require a fixed definite processing temperature. Such fixed temperatures occur in the curing of rubber, the moulding of plastics, etc. Having fixed the processing temperature, it is then necessary to find out how small need be the temperature difference between the heating steam and the process material. If the heating is done by direct injection, there may even be no need for a temperature difference at all. (There are even cases, in sugar or caustic solutions for example, where steam injected into the solution can raise the temperature of the solution above its own steam temperature. This is not

contrary to the Second Law of Thermodynamics and is discussed in more detail in Chapter 25.) If the heat is supplied through a heating surface, it may be possible to reduce the temperature difference by increasing the heating surface, by increasing the movement of the material, by circulating the steam, by ensuring that the steam is free from air or gas, that the condensate is properly drained, etc. These things are discussed in other Chapters.

When the processing steam pressures for all parts of the plant have been decided upon it will probably be found that a considerable range of pressures is required. It is necessary to estimate or measure the steam consumption at the various pressures. Assume that the following quantities of steam are needed at the following pressures, and that the boilers are in good condition and do not warrant renewal and that their working pressure is 200 psi.g. :—

Steam	*Pressure*
1,000 lb./hr.	50 psi.g.
5,000 ,,	20 ,,
5,000 ,,	5 ,,

If we select 50 psi as the exhaust pressure the whole 11,000 lb. of steam per hour can be passed through the engine. If we select 20 psi as the exhaust pressure 10,000 lb. can be used in the engine, the 50 psi steam being blown direct through a reducing valve. Reference to Table X, Section 104, and a little interpolation enables us to tabulate thus :—

Exhaust pressure	*Steam available for engine*	*Power obtainable (reciprocator)*
50 psi.g.	11,000 lb./hr.	260 H.P.
20 ,,	10,000 ,,	361 ,,
5 ,,	5,000 ,,	207 ,,

It is obvious that we should choose 20 psi as the back pressure in this simple example. This will mean that the 50 psi steam must be reduced direct from the 200 psi main through a reducing valve, and the 5 psi steam must be reduced from the 20 psi back pressure main, or better, through some piece of plant which will do useful work.

126. BLOWING OFF EXCESS EXHAUST STEAM. The second method of bridging the power gap listed in Section 124 is blowing off the excess exhaust steam to atmosphere. This may appear at first sight to be very wasteful. Of course it is ; but it is not necessarily so wasteful as it appears and may not be so wasteful as some of the other possibilities.

Look again at Fig. 38. The no-load steam consumption of this engine is 3,700 lb./hr., most of it passing straight through the engine. If the process requires 3,700 lb./hr. then any power generated by the engine will call for 17 lb./H.P./hr. which must be blown-off to atmosphere. On the face of it the engine uses about 30 lb. of steam per H.P./hr., but the actual amount of steam to be blown away is only 17 lb./H.P./hr. provided the process is using the no-load steam from the engine. So long as the exhaust steam is being properly used most of the time for process or heating, it may be more economical to blow off occasionally than to abandon back pressure generation.

As an economical back pressure machine, this engine is probably using $3{,}415 \times 1\cdot18$ Btu/kWh.

If the boiler house efficiency is 70 per cent. and the coal contains 12,000 Btu/lb., the coal consumption will be

$$\frac{3{,}415 \times 1\cdot18}{12{,}000 \times \cdot7} = \cdot48 \text{ lb./kWh.}$$

If, to generate extra power, 17 lb. of steam per H.P. or $22\cdot8$ lb./kWh have to be blown off, the whole of the heat in this blown-off steam will be lost.

The coal consumption under these conditions, for the blown-off steam, with the same boiler house efficiency, will be :—

$$\frac{22\cdot8 \times 1{,}170}{12{,}000 \times \cdot7} = 3\cdot18 \text{ lb./kWh.}$$

Now the average grid coal consumption is $1\cdot4$ lb./kWh. We can find how much power it is Nationally economical to generate by blowing off thus :—

Let x = the fraction of blow-off power.

$$\begin{aligned} \text{Then}\quad 3\cdot18x + (1 - x)\cdot48 &= 1\cdot4 \\ 2\cdot7x &= \cdot92 \\ x &= \cdot34 \end{aligned}$$

Therefore one-third of the power can be generated by blowing off, with this particular engine, and the other two-thirds by back pressure working in order that the coal used will just equal average grid consumption.

By equating costs instead of coal, the amount of power that it *pays* to generate by blowing off can be found.

127. SEPARATE CONDENSING SET. If the increased power demand, which cannot be met, is due to an increased electrical load, so that the existing set is fully loaded, then the decision lies between a separate condensing set and one of the other methods, and the argument is fairly simple. If, however, the shortage of power is due to a saving in process steam demand, then things are not quite so straightforward. If we consider the plant to be of the same size as we have considered in Section 121, then the additional plant that is being considered will probably be of less than 200 H.P.

An engine of this size taking steam at 200 psi will use something over 20 lb. of steam per H.P./hr., when exhausting to a condenser.

But we have seen in Section 126 above that by blowing off we shall only waste 17 lb. of steam per H.P. So clearly it is not only more economical in this case to blow off, but we save the cost of the new set. The reason for this apparently strange state of affairs is that with the back pressure set we are only blowing off marginal steam, whereas in the new small condensing set we have to use both marginal and fixed steam.

128. BUYING FROM THE PUBLIC SUPPLY. From the National point of view it is better to purchase current from the grid than to blow off or to use a small condensing set. If the excess load is a constant load, the grid can always

beat the coal consumption of non-back pressure industry. If the load coincides with grid peaks the advantage of the grid may disappear.

It very often pays to generate part of the electrical demand and purchase the remainder from the public supply. Unfortunately difficulties often arise when an attempt is made to run in parallel with the grid. These are discussed in Section 804. The result of these difficulties is to encourage the purchasing of the whole of the power requirements. This is extravagant and wrong from the National point of view, as far as coal consumption is concerned, if the factory has a boiler plant. Another choice is to generate the whole of the power by means of the factory back pressure plant blowing off the surplus exhaust that the process cannot use. This is also extravagant but it may not be nearly so extravagant as the complete abandonment of back pressure power generation.

A compromise is to split the electrical circuits into two and run one section on the public supply and the other from the factory power plant.

129. ENSURING THE MINIMUM BLOW-OFF. If steam is to be blown off for short periods to allow extra power to be generated over and above that which can be produced from the process steam, it should be blown off deliberately and not simply left to find its own way out by some safety valve. In the case of the engine whose performance figures are given in Fig. 38, the inlet pressure is 200 psi and the back pressure is 20 psi.g. The safety valve on the 20 psi line is probably set to blow at 25 psi.g. If the engine back pressure is allowed to rise to 25 psi.g. the steam consumption of the engine will increase from 17 lb./H.P. to the equivalent of say 18½ lb./H.P., due to the increase in the fixed steam needed to overcome the higher back pressure. When it is known, therefore, that blow-off must occur, the back pressure should be kept down so that the amount of steam to be blown away will be kept to the minimum. It goes against the grain with many engine drivers to blow steam to atmosphere deliberately when, by leaving it to the safety valve, which has not yet opened, this steam would seem to be saved. But leaving it to the safety valve ensures that more than is necessary will eventually be blown away. The best method of securing the desired result is to have two valves. The normal safety valve, on which the safety of the plant depends, and a second relief valve which is set about 5 psi lower and can be brought into use by opening a stop valve in front of it.

The cumulative evil of increasing back pressure should always be borne in mind. Let us say that an engine or turbine is running on a steady load and exhausting to a steady process at a satisfactory back pressure. If now a part of the process is shut down, the back pressure rises ; this calls for more steam for the engine, which makes the back pressure even higher, and so the vicious circle goes on. While this is happening the other end takes a hand. The increased governor opening slightly lowers the boiler pressure so that to get the necessary power the governor opens still wider. Many back pressure factories have experienced the maddening state of affairs when the boiler pressure is low, the engines are flat out, low pressure safety valves are roaring and still the needed power is not being produced.

(87684)

130. PASS-OUT MACHINES. A pass-out machine can be either a compound or triple expansion reciprocator or a turbine. The machine is connected at its exhaust end with a condenser. At some suitable point in the expansion (in a turbine) or between the H.P. and L.P. cylinders (in a reciprocator) a connection is brought out which passes out the necessary steam to the process. The effect of this is to combine in one machine a back pressure set and a condensing set. It might be thought that this would greatly reduce the losses by concentrating them all in one machine. There is, however, a new loss. If a back pressure set provides most of the power and is supplemented at peaks by a condensing set, the latter can be shut down and its losses cut off when the electrical demand is small enough to be met by the back pressure set alone. In a pass-out turbine some steam must always be passed through to the condenser in order to keep the low pressure end of the turbine *cool*. This may sound surprising, but if the low pressure end were allowed to revolve in stagnant vapour, the churning by the blades would soon raise the temperature to a dangerous level. In the same way some steam must be passed through the low pressure cylinder of a pass-out reciprocator lest it become a pump. The result of this at light loads is unfortunate. Some steam, passed through the low pressure end, must be condensed and its latent heat lost. This steam, undergoing the full pressure drop, generates a relatively large amount of power. This results in there being less available to pass out to the process, which may have to call on the reducing valve to make up the shortage direct from the high pressure main.

A pass-out machine is not therefore necessarily the ideal machine if the excess power demand is only rare. The condensing pass-out machine has its best application where the process always, or almost always, calls for considerably less steam than is necessary to produce the power load. A pass-out turbine will work best if the pass-out quantity is relatively steady and especially if the pass-out pressure can be allowed to vary. In many ways a reciprocating engine makes a better pass-out machine than does a turbine. A tandem compound engine is much more adaptable to pass-out working than a cross compound. This point is discussed again in Section 156 below.

131. BACK PRESSURE PASS-OUT AND MULTIPLE PASS-OUT SYSTEMS. Where the process calls for several pressures a decision has to be made as to the best back pressure to select. This was discussed in Section 125. In the example given in Section 125 the process called for three different pressures. It was decided that the best back pressure to use for a straight back pressure set was the intermediate process pressure, the low pressure supply being reduced from the back pressure main and the high process pressure being reduced from the high pressure main. This example is rather too small a plant for the application of a multiple pass-out turbine and the pressures are not suitable for the use of a triple expansion engine, but a compound engine might give good results if steam were passed out between the high and the low pressure cylinders.

The steam requirements to process are repeated at the top of the next page.

We should put the whole 11,000 lb. into the engine at 200 psi and the H.P. cylinder would exhaust at 50 psi. 6,000 lb. would be passed out at 50 psi and

Quantity, lb./hr.	*Pressure, psi.g.*
1,000	50
5,000	20
5,000	5
11,000	

5,000 of this pass-out steam would be reduced to 20 psi. The remaining 5,000 lb. would go into the low pressure cylinder and would exhaust at 5 psi.

With 11,000 lb. of steam per hour we find from Table X that we can expect an efficiency ratio of about 64 per cent between 200 and 50 psi,

and that the ideal engine horse power will be $375 \times 1 \cdot 1 = 413$ H.P.

So that we can expect to get 264 H.P. from the high pressure cylinder.

5,000 lb. of steam will go through the low pressure cylinder from which we can expect an efficiency ratio of about 55 per cent.

We must calculate the heat drop over the low pressure cylinder and must first find the state of the steam at the high pressure cylinder exhaust.

The total heat of 200 psi.g. saturated steam is 1,200 Btu.

We have assumed that we shall get 264 H.P. from 11,000 lb./hr. of steam

so that the real heat drop will be $\dfrac{264 \times 2{,}545}{11{,}000} = 61$ Btu/lb.

The exhaust from the high pressure cylinder will therefore contain

$$1{,}200 - 61 = 1{,}139 \text{ Btu/lb.}$$

The temperature of 50 psi.g. steam is 298° F. at saturation

and Gibbs' Function = 62,

so the entropy at exhaust will be $\dfrac{1{,}139 + 62}{298 + 460} = 1 \cdot 584$.

The temperature of the 5 psi.g. steam exhausting from the low pressure cylinder is 227° F. and " G " = 35.

So that the total heat in the low pressure exhaust will be

$$(227 + 460)\ 1 \cdot 584 - 35 = 1{,}053 \text{ Btu.}$$

The adiabatic heat drop over the low pressure cylinder is $1{,}139 - 1{,}053 = 86$.

At an efficiency ratio of 55 per cent. the actual heat drop will be

$$86 \times \cdot 55 = 47 \text{ Btu.}$$

The expected low pressure horse power will be $\dfrac{47 \times 5{,}000}{2{,}545} = 92$ H.P.

So the total horse power from both cylinders will be $264 + 92 = 356$,

instead of 361 with the straight back pressure engine exhausting at 20 psi. This shows virtually no gain over straight back pressure working.

This calculation is an over-simplification. It should be corrected thus :—

The entropy of saturated 50 psi.g. steam is 1·638.

The entropy of evaporation is 1·204.

The wetness of the H.P. cylinder exhaust is $\frac{1\cdot638 - 1\cdot584}{1\cdot204} = 4\cdot49$ per cent.

For every 100 lb. of dry saturated steam required at the H.P. cylinder exhaust, we shall have to put in 104·7 lb. of 200 psi steam into the engine.

The total heat in the 5 psi exhaust is 1,053 and the total heat at 5 psi saturation is 1,156, the latent heat being 961.

The wetness in the 5 psi exhaust (assuming the moisture in the H.P. exhaust was removed in a separator) will be $\frac{1{,}156 - 1{,}053}{961} = 10\cdot7$.

To get 5,000 lb. of 5 psi process steam from the L.P. exhaust we must put 5,600 lb. of 50 psi steam into the L.P. cylinder.

To get 1,000 lb. of 50 psi steam for process, 5,600 for L.P. cylinder and 5,000 for 20 psi process will call for $(1{,}000 + 5{,}600 + 5{,}000)\ 104\cdot7 = 12{,}145$ lb. into the H.P. cylinder.

We shall therefore get $\frac{264 \times 12{,}145}{11{,}000} = 292$ horse-power from the H.P. cylinder and $\frac{92 \times 5{,}600}{5{,}000} = 103$ from the L.P. cylinder—a total of 395 horse-power for an extra steam input of 1,145 lb./hr.

The straight back pressure engine must be similarly corrected for exhaust moisture. We shall need to put in 10,525 lb. instead of 10,000 lb. and will get 380 horse-power instead of 361. This shows the pass-out machine as slightly better than the straight back pressure engine.

Let us multiply this factory by ten so that conditions are suitable for a multiple pass-out back pressure turbine and make the comparison against straight back pressure again. In the multiple pass-out back pressure machine 110,000 lb. of steam will be put into the turbine and 10,000 lb. will be passed out at 50 psi.g., 50,000 will be passed out at 20 psi.g. and 50,000 will go through to the exhaust at 5 psi.g. The straight back pressure set at 200 to 20 psi, and the pass-out increments can be conveniently tabulated thus :—

	Quantity lb./hour	*Inlet pressure psi.g.*	*Exhaust pressure psi.g.*	*Adiabatic heat drop Btu/lb.*	*Adiabatic horse-power*	*Effici-ency ratio*	*Actual horse-power*
Straight back-pressure	100,000	200	20	155	6,090	71	4,320
Multiple pass-out back-pressure	110,000	200	50	119	4,930	70	3,450
	100,000	50	20	56	2,200	68	1,500
	50,000	20	5	44	860	64	550
							5,500

This gives the pass-out machine an advantage of 27 per cent. over the straight back pressure set.

In the large back pressure and multiple pass-out turbines the steam has been given sufficient superheat to give dry saturated exhaust. So that, apart from gland leakage, the steam passed out plus the exhaust steam will equal the steam put in.

It is impossible to dogmatise as to when a pass-out condensing, a pass-out back pressure, a multiple pass-out or a straight back pressure set should be used. Each case must be gone into individually. A word of caution is, however, not out of place. Process demands have a nasty way of changing both as to pressure and quantity, up, down, high, low. Electrical load also has a horrible tendency to go up, up, up. Any very saucy multiple pass-out machine may fill the bill to-day and be quite unsuitable tomorrow.

The costing of pass-out power is anything but straightforward. It is dealt with in Chapter 21.

A pass-out machine when working with fluctuating demand for power and varying quantities of pass-out steam is not so efficient as a straight back pressure set, and its regulation, if the pass-out pressures are to be kept reasonably constant under fluctuating electrical load, is somewhat complicated. In many industries, for example the paper industry, pass-out machines give excellent service, and there are many factories where reciprocating engines can work as pass-out machines extremely well.

132. EXHAUST TURBINES. An exhaust turbine works in conjunction with a back pressure engine or turbine, or it can be installed to take the exhaust steam from engines, turbines, pumps, hammers, etc., that have hitherto been exhausting to atmosphere. There is no need to argue the obvious benefits from its use in the latter cases. It is only necessary to give two examples, small and large, and then the merits or faults of the exhaust turbine over the pass-out condensing or separate condensing sets can be discussed.

Let us take the cases of exhaust turbines put in to use the exhaust steam from machines that have hitherto been blown to atmosphere. The first, a relatively small set taking 12,500 lb. of steam/hour ; the second, a larger machine taking 125,000 lb./hr. In the first case we shall assume a condenser vacuum of 28 in. and in the second 29 in.

Steam quantity	*Inlet pressure*	*Exhaust pressure*	*Adiabatic heat drop*	*Efficiency ratio*	*H.P.*
12,500	Atm.	28 in.	170	68	570
125,000	Atm.	29 in.	204	80	8,020

These figures should not surprise us after our study of Section 82, but they are a never-ending source of wonder to the author. Clearly every possible effort should be made to use atmospheric steam whether it be the exhaust from an engine or the exhaust from an evaporator or pan. There are difficulties due to intermittent production and there may not be a use for the power, but the benefits are so striking that all efforts should be made by hook or crook to find a use or a customer for the power and to get over any technical difficulties.

Let us now consider a process factory with a back pressure set and an exhaust condensing turbine. First assume a constant process steam demand which is slightly less than the steam called for by the electrical load. All the

steam will be going through the back pressure set and a small amount through the exhaust turbine. If there is an increase in electrical demand the back pressure set calls for more steam, it generates more power and makes more exhaust which passes through the exhaust turbine which also generates more power. There may be a slight lag in response and possibly a tendency to hunt and the governing arrangements require careful design.

Now suppose the electrical demand is constant at normal load and the steam demand for process varies. When the process demand is zero the exhaust turbine will be well loaded and the back pressure set will be generating a relatively small proportion of the power and the whole of the back pressure exhaust goes into the exhaust turbine. As the process demand increases so does more and more steam go through the back pressure set until, if the electrical load is not too great, the back pressure set is meeting the whole electrical demand. The exhaust turbine, however, is still taking its no-load steam consumption, in order to run at full speed and make up its losses. At this point, if conditions are likely to remain steady, the exhaust turbine would be shut down and any excess exhaust from the back pressure set blown to atmosphere. Any further increase in process demand would absorb this blow-off.

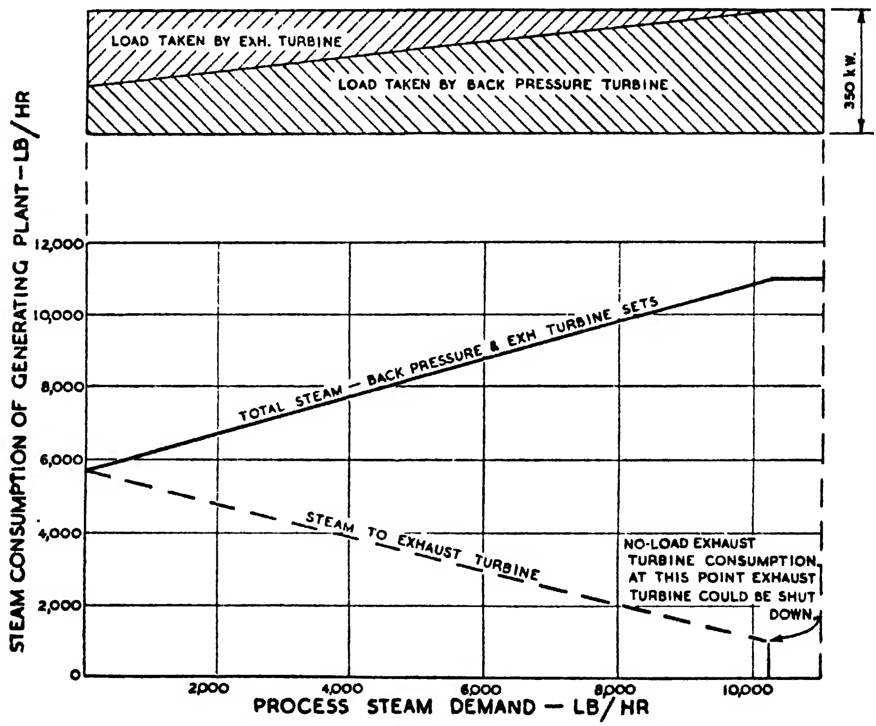


FIG. 41. STEAM CONSUMPTION OF BACK PRESSURE ENGINE AND EXHAUST TURBINE WITH CONSTANT ELECTRICAL LOAD AND VARYING PROCESS STEAM DEMAND

Fig. 41 shows how a constant electrical load of 350 kW would be shared by a back pressure set and an exhaust turbine over the range from no process steam to process absorbing the whole of the back pressure exhaust. In this example,

high pressure steam at 350 psi and 600° F. is supplied to the engine which exhausts into a process main at 20 psi.g. The exhaust turbine has been assumed to have a no-load consumption of 1,000 lb./hr.

The exhaust turbine in combination with a back pressure set is generally more efficient than a pass-out condensing turbine. On light loads it has the advantage that it can be shut down whereas the pass-out set must put some steam through the low pressure end to cool it. Again, however, every case must be judged on its own particular conditions and requirements.

133. MIXED PRESSURE TURBINES. The mixed pressure turbine is sometimes a useful solution to the power difficulty. These turbines are like inverted pass-out turbines; they receive steam at more than one pressure. In Fig. 42 on page 110 the example shown at H is a mixed pressure turbine taking its main supply at 200 psi and receiving into its low pressure end exhaust steam at atmospheric pressure as well. Like pass-out turbines, governing and control sometimes present difficulties. Theoretically a mixed pressure turbine working in conjunction with engine exhaust should not only be economical but should even out the demand on the boilers. When the engine load is added to the power load, instead of the whole engine load being additional, all the engine exhaust passes into the mixed pressure turbine and relieves it of much of its demand for live steam. There are unfortunately other factors. When no exhaust steam is being taken, the low pressure end is not properly loaded. When much exhaust steam is being taken the high pressure end is not properly loaded. The mixed pressure turbine is more poorly placed in this respect than the pass-out turbine as there can seldom be any condition of steam supply in which the whole turbine is properly loaded, whereas it is possible to work a pass-out turbine in such a way as to load every stage to full capacity most of the time. Since the mixed pressure turbine is generally badly loaded, it always tends to be too big a machine and is usually less efficient than a separate exhaust turbine working in conjunction with a straight back pressure machine.

Mixed pressure turbines are in extensive use and often give very satisfactory results, particularly in the Coal Mining and Steel industries.

134. VACUUM TURBINES. Many industries, caustic recovery, sugar refining, milk condensing, etc., reject their process heat in the form of vapour at about 24 in. vacuum. This vapour is usually condensed in jet condensers. Such vapour will, in a fairly large machine, give 1 electrical unit for 40 lb. of steam, if it is put into a vacuum turbine exhausting into a surface condenser at 29 in. vacuum. 100,000 lb. of such steam per hour will give 2,500 kW. This is even more striking than the output from a turbine taking steam at atmospheric pressure.

The total heat of steam at 24 in. vacuum is 1,123 Btu.

The entropy is 1·889.

The temperature of 29 in. vacuum steam is 79° F. and $G = 2$.

The total heat after adiabatic expansion from 24 in. to 29 in. will be

$$(79 + 460)\ 1{\cdot}889 - 2 = 1{,}018.$$

So that the adiabatic heat drop will be 1,123 — 1,018 = 105 Btu.

Suppose it be assumed that there is an output of 100,000 lb. of 24 in. steam per hour, the ideal horse power will be $\frac{100{,}000 \times 105}{2{,}545} = 4{,}125$.

Table IX shows that we can expect an efficiency ratio of 78 per cent., so that we can reasonably expect to generate 2,400 kW.

Take the case of a much smaller plant, say one that is rejecting 10,000 lb. vapour per hour at 24 in. vacuum. We will assume that the smaller plant will only be able to operate its condenser at 28 in. vacuum.

This gives an adiabatic heat drop of 68 Btu.

If the turbine has an efficiency ratio of 67 per cent. it will give

$$\frac{10{,}000 \times 68 \times \cdot 67}{3{,}415} = 133 \text{ kW.}$$

The disadvantage of these machines is their high cost. They are very large machines—that is why they are efficient. The " little " machine just considered would probably have a single row, or at most two rows, of blades and these blades and the condenser would have to be the same size as on a 1,000 kW. straight condensing machine. Another handicap is that to get very high vacuum calls for a great deal of cold cooling water, and the pumping load for this water must be deducted from the gross output of the vacuum turbine to get its net output. Apart from maintenance cost and capital cost which admittedly is very high, the power is obtained absolutely free.

135. HEAT ACCUMULATION. We are concerned not only with total steam demand and total electrical load, but with minute to minute variations. It often happens that the total steam demand from the process is sufficient to enable the total power demand to be met, but only too often light process steam demand coincides with heavy electrical demand and vice versa. These conditions call for momentary blowing off of excess exhaust steam, or for short-circuiting of the power plant through the reducing valve. Some form of heat accumulator may enable the steam that would have been blown off to be stored so as to meet any later demand that might call for steam through the reducing valve.

It has been shown in Section 132 that an enormous amount of power could be obtained from steam at atmospheric pressure by passing it through an exhaust turbine. In many works there are engines exhausting to atmosphere for short sharp periods—for example, the steam hammers and rolling mill engines in a steel works. If this steam could be collected and the peaks smoothed out a great deal of power could be generated in an exhaust turbine.

There is only one way of storing exhaust steam at absolutely constant pressure and this is rarely used. Most accumulators store at varying pressures and, unless they are large, the pressure variation may be too great for this particular application.

The standard form of exhaust steam accumulator is simply a large tank half full of water and capable of withstanding a pressure of about 15 psi. There are virtually no controls, except a safety valve. The exhaust steam is blown into the water through a row of blowers. When the exhaust steam user takes less steam

than the supply, the exhaust back pressure rises ; this raises the boiling point of the water in the accumulator and the steam condenses in the water. When the exhaust steam user calls for more steam than is being exhausted, the back pressure is lowered, the boiling point of the water in the accumulator drops and the accumulated steam flashes off.

The capacity of such an accumulator is small. For example, an accumulator constructed from an old 30 ft. × 8 ft. 6 in. Lancashire boiler will only store 1,000 lb. of steam with an overall pressure drop of 5 psi. But 1,000 lb. of steam stored in a minute and discharged during the next two minutes will, on a regular cycle, handle 20,000 lb. steam an hour.

In an accumulator the blower nozzles must be well submerged—say by 1 ft. This imposes a permanent additional pressure drop of 0·5 psi on the system.

The storage capacity of the accumulator can be ascertained thus :—

Total heat in steam at max. pressure (say 5 psi.g.) is 1,156·3 Btu./lb.

Sensible heat in water at max. pressure (say 5 psi.g.) is 195·5 Btu./lb.

Sensible heat in water at min. pressure (say 0 psi.g.) is 180·2 Btu./lb.

Heat capacity of water from 0 psi.g. to 5 psi.g. is 195·5–180·2 or 15·3 Btu./lb.

Amount of water needed per 1 lb. steam storage is

$$\frac{1{,}156 \cdot 3}{15 \cdot 3} \text{ or } 75 \cdot 5 \text{ lb. or } 1 \cdot 26 \text{ cu. ft.}$$

It may be very important to reduce the pressure drop to a minimum, but this greatly increases the size of the accumulator. Table XV shows what can be obtained.

TABLE XV. EXHAUST STEAM ACCUMULATORS

NET PRESSURE DROP IN ACCUMULATOR PSI.G.	GROSS PRESSURE DROP OVER PLANT PSI (HYDROSTATIC HEAD ·5 PSI)	LB. WATER PER LB. STEAM STORED	WATER CAPACITY NEEDED TO STORE 1,000 LB. STEAM CU. FT.	EQUIVALENT NO. OF LANCS. BOILER SHELLS (30′ × 8′ 6″)
1 to 0	2	340	5,460	6
2 to 0	3	174	2,790	3
3 to 0	4	120	1,930	2
4 to 0	5	92	1,475	1⅔
5 to 0	6	75	1,205	1⅓

Steam accumulators are dealt with in detail in Chapter 16.

136. FEED WATER HEATING. If steam is being blown to atmosphere or its latent heat lost in a condenser, it always pays to use steam to heat the boiler feed water.

Section 88 explains the great improvement in efficiency obtained by regenerative feed heating. It explains that by this means the power station is doing a little bit of pass-out process work.

As the complete back pressure cycle is 100 per cent. efficient, there is no virtue in using exhaust or bled steam for feed heating because all the exhaust heat is being usefully employed. If, however, some steam is exhausting to atmosphere or to a condenser, it pays to do as much stage feed heating—power station fashion—as will reduce the wasted exhaust steam as much as possible.

Where new back pressure or pass-out plant is being installed, bleeder points can be arranged for on the turbine. Where existing plants are concerned we have to take what there is. There are sometimes drainage connections or dirt boxes at points along a turbine casing, and the makers should be consulted as to whether these could be used as bleed points. With a back pressure or pass-out reciprocator, two stage feed heating is generally all that is possible—from the exhaust and from the low pressure receiver.

In the planning of one large factory it was found possible to increase the power output from 8,500 kW to over 10,000 kW by carrying feed water heating to the limit.

Hotter feed water may slightly reduce the economiser efficiency, but the efficiency of the whole plant will be so increased that such slight falling off in boiler efficiency is probably of little account.

137. SELLING THE SURPLUS EXHAUST STEAM. Far better use can be made of the heat in excess exhaust steam from a back pressure set than by wasting it in the condenser of a pass-out set or exhaust turbine if it can be supplied to a neighbouring factory. The difficulties are of three kinds, practical, psychological and financial.

From the practical point of view the supply must be steady in both pressure and quantity or it will be of no use to the neighbour. For this reason it is generally useless to try to sell peak load steam. Sometimes a supply at night only is convenient to both parties. Whatever the difficulties they should if possible be surmounted, because waste steam, used by a neighbour, is pure net gain.

There are two serious psychological difficulties. The receiver of steam does not like the feeling that he is dependent for his supply on " those beggars next door " whom for years he has probably despised. The steam supplier does not like the feeling that he has an exhaust obligation to fulfil. There is no reason why a supply from a private neighbour need be any more uncertain than a supply from a public undertaking ; in fact many factories pride themselves that they are as reliable as any public supplier. If the neighbourly supply is replacing an existing supply, the existing plant is an adequate spare. Goodwill and liaison should be able to circumvent psychology.

The financial arrangement is usually the principal stumbling block. The recipient of steam objects, not unnaturally, to being asked to pay for steam that he has watched for years defiling the local heavens. Unfortunately, the surplus steam suppliers' accountant turns greedy eyes on his neighbour's smoking chimney.

The metering of neighbourly steam should be done by steam meter, preferably in the receiver's works. Measurement by condensate may unduly favour the supplier if the steam was wet.

Losses in transmission are sometimes brought as an argument against such a steam link. The losses in a really well lagged pipe are surprisingly low. As to how far steam can be sent, no general statement can be made. The greater the quantity, the lower the proportional losses ; consequently the further the steam can be sent. After enough steam has come out at the far end of the pipe to pay for the cost of the pipe in a reasonable time, the rest is pure gain.

138. LONG DISTANCE STEAM TRANSMISSION. Three actual examples of heat losses on long steam pipes are shown below.

These results are moderately consistent. The small high pressure pipe, not too well-lagged, shows the biggest loss. The large low pressure well-lagged pipe shows a very small loss. But in all three cases the losses are really very low and indicate that steam can be carried long distances without undue loss, and often with great saving in other directions.

Between two Liverpool factories

Pipe diameter	15 in.
Length	1,470 ft.
Lagging	3 in. magnesia, ½ in. hard compo.
Capacity	40,000 lb./hr.
Pressure	40 psi.g.
Heat loss	·3 Btu/sq. ft./° F/hour *or* 404 Btu/foot run/hour *or* 1·6 per cent. of latent heat at full load *or* ·11 per cent. of latent heat per 100 ft.

Between two Leicester factories

	A.	*B.*
Pipe diameter ..	3½ in.	9 in.
Length	2,100 ft.	2,100 ft.
Lagging	2 in. plastic magnesia ½ in. hard compo.	2½ in. plastic magnesia ½ in. hard compo.
Capacity	5,000 lb./hr.	20,000 lb./hr.
Pressure	110 psi.g.	60 psi.g.
Heat loss	·6 Btu/sq. ft./° F/hr.	·4 Btu/sq. ft./° F/hr.
or		
	278 Btu/foot run/hour	305 Btu/foot run/hour
or		
	13 per cent. of latent heat at full load	3·5 per cent. of latent heat at full load
or		
	·63 per cent. of latent heat per 100 ft.	·17 per cent. of latent heat per 100 ft.

139. RUNNING A POWER ECONOMY CAMPAIGN. If we can prevent the problem of excess power demand arising or if we can eliminate it once it has poked up its ugly head, there will be no need to consider all kinds of fancy arrangements for meeting it. The way is clear. Use less power. It can be stated quite definitely that waste of power is just as widespread and is just as bad as waste of steam. As the operation of a power-saving campaign is hardly power generation technique, it will not be discussed here. It is dealt with in Chapter 22.

140. SELLING POWER. We have so far been dealing with factories that have difficulty in generating sufficient power. There remain the factories in that category where the steam consumption is so great that the power requirements can be met—and more.

Such factories should, of course, generate all the power they can and should sell the surplus to the local electricity company. Even small factories, provided they use all their exhaust steam, can generate power more economically than a big power station.

There are sometimes objections to parallel or synchronous running between industrial plants and the Grid. There are certain ways in which these objections can be met—they are discussed in Section 804. Every effort should be made to smooth out the difficulties that selling power presents. It is very profitable and greatly in the national interest.

141. DESIGN FACTORS. In designing a plant there are many factors ; among the most important are :—

(*a*) The choice of back or pass-out pressures.
(*b*) The choice of boiler pressure.
(*c*) The choice of high pressure or high superheat.
(*d*) The choice of machine type.

The first has been discussed already in Section 125 ; the other three must now be considered.

142. CHOICE OF BOILER PRESSURE. This choice is only open when installing new plant. If old plant is being converted to back pressure working, all that can be done is to work the boilers at the highest possible pressure for which they can be insured, and to reduce the back pressure to the lowest possible point. Even with a perfectly free hand the choice of best pressure and temperature is difficult. A few examples will show just what are the considerations involved.

Boiler pressures must be reasonably low in small plants, as the gain in efficiency by going to very high pressures in a very small plant are usually not worth the extra complications.

There are plants in operation at very high pressures, and if it is very important that the very last kW should be generated, it may be worth considering very advanced plant. But high pressure plant has a nasty knack of becoming the master instead of remaining the servant. The essential need of a process factory is the correct operation of the process, and a very high-brow power plant leads to a serious diversion of technical and managerial effort.

The power stations must simply try to produce power in the most economical way and it should be the power station's task and privilege to lead the way in improved efficiencies.

We will now investigate three cases and try to see what the problem is.

143. CONDITIONS IN A SMALL DYE WORKS. Assume that the plant requires 10,000 lb. of steam at 20 psi.g. (this is an over-simplification, but were the real conditions taken the argument might be wearisome). We can take 250 psi as a nice prudent pressure. We can get this pressure on a shell boiler which may be a great advantage ; to go higher may mean going to a water-tube boiler.

Table X tells us that we can expect an efficiency ratio of 65 per cent. from an engine taking 10,000 lb. of saturated steam/hour exhausting at 20 psi.g. and that we should produce 65 per cent. of 610 H.P. = 396 H.P. = 295 kW.

The turbine efficiency ratio is given as 49 per cent. This could be improved by superheat but not sufficiently to bring the turbine up to the engine. The engine efficiency could also be improved by superheat, but this will call for lavish cylinder lubrication, and we know that the steam is going to come into contact with our future shirts, so the lubrication necessary with superheat is probably out of the question.

295 kW represents the most power we can hope to generate without doing something that may not be satisfactory as regards trouble-free operation and diversion of effort. The dyer's business is dyeing—not operating miniature "Batterseas". If more power than 295 kW is needed, the balance should be purchased. (See Section 804.)

144. CONDITIONS IN A PAPER MILL. We will assume that the process demand is 100,000 lb. steam per hour at 40 psi.g. and that the electrical load is 5,000 kW. This calls for 20 lb. steam per kWh.

Table VIII tells us that 1 kWh = 3,415 Btu, so the actual heat drop must be

$$\frac{3{,}415}{20} = 171 \text{ Btu.}$$

To get such a heat drop with 40 psi exhaust will call for an initial pressure of well over 400 psi.

From Table IX we can assume an efficiency ratio of about 70 per cent.

The adiabatic heat drop will therefore be $\frac{171}{\cdot 7} = 244$ Btu.

The exhaust must have a little superheat, say 50°, to ensure reasonable dryness at the process plant. The steam table tells us that 40 psi.g. steam at 340° F. will have 53° of superheat and contains 1,204 Btu.

This then is to be the real exhaust state.

The adiabatic exhaust steam would contain 244 − 171 = 73 Btu less heat, or a total of 1,131 Btu.

We must find the entropy of 40 psi steam containing 1,131 Btu.

The temperature of saturated steam at 40 psi is 287° F. and Gibbs' Function is 57.

The entropy will be $\frac{1{,}131 + 57}{287 + 460} = 1 \cdot 590$.

The initial high pressure steam must therefore have an entropy of 1·590 also.

And its total heat must be $1{,}131 + 244 = 1{,}375$ Btu.

From the superheated steam table it is found that the only steam having these properties is at a pressure of 700 psi.g. and a temperature of 750° F.

The same result can be obtained much more quickly by using the Mollier chart. The desired entropy is found from the intersection of the 40 psi.g. pressure line on the 1,131 Btu ordinate. Then the intersection of 1,375 Btu and 1·590 entropy gives the state of the initial steam.

Plant at this pressure and temperature may be more advanced than we are prepared to instal. The only way out of the difficulty will be either to buy extra power over that which a lower pressure back pressure plant will give, or to use as high a pressure as we think would be trouble-free and to employ a pass-out condensing turbine. It will almost certainly pay in £.s.d. to use the pass-out turbine instead of buying power, but will probably be Nationally wasteful.

(When doing problems of this kind without the help of a Mollier or Temperature-Entropy diagram, it is always helpful to draw rough sketches of either or both diagrams to keep one on the right lines.)

145. CONDITIONS IN A SUGAR REFINERY. A sugar refinery is planning a new power plant. It is at present using 1·5 tons of steam per ton of sugar and its power load is 55 kWh per ton of sugar. It estimates that steam savings can be effected sufficient to bring the steam consumption down to 1·1 tons/ton and it estimates that this will be accompanied by an increase in power load up to 65 kWh per ton. To insure the future, it decides to instal the new power plant, if practicable, on the basis that the steam demand may be reduced to ·9 ton/ton (this being "bogey"—see Chapter 20) and the power demand increased to 72 kWh per ton. It has been decided that the process pressure cannot be reduced below 60 psi.g. and that the exhaust must leave the turbine with about 40° F. of superheat. The weekly output is 12,000 tons and the working week is 144 hours.

The steam demand will be approximately

$$\frac{12{,}000 \times \cdot 9}{144} \text{ tons} = 168{,}000 \text{ lb./hour.}$$

The electrical load will be approximately $\frac{12{,}000 \times 72}{144} = 6{,}000$ kW.

This means that the plant must be designed to produce 1 kW for 28 lb. steam.

The net heat drop of the cycle is $\frac{3{,}415}{28} = 122$.

The load is to be taken by two 3,000-kW turbines. Table IX shows that from a 4,000-H.P. machine working over the kind of range that will obviously be needed we can expect an efficiency ratio of about 69 per cent.

The adiabatic heat drop will be $\frac{122}{\cdot 69} = 177$ Btu.

Steam at 60 psi.g. (75 psi.a.) and 350° F. has a superheat of 43° F. and this will be a satisfactory exhaust-to-process condition.

Such steam contains 1,206 Btu.

Now the net heat drop is to be 122 and the adiabatic heat drop is expected to be 177.

The adiabatic exhaust will have a heat content of $177 - 122 = 55$ Btu lower than the actual exhaust state.

The adiabatic exhaust heat content will be $1{,}206 - 55 = 1{,}151$ Btu.

The temperature of 60 psi.g. saturated steam is 307° F.

And its total heat is 1,182 Btu.

The adiabatic exhaust is therefore wet so we can use Gibbs' Function to calculate the adiabatic entropy.

The value of "G" at 60 psi.g. is 66 Btu.

The entropy will be $\frac{1{,}151 + 66}{307 + 460} = 1 \cdot 587$.

The actual exhaust has a heat content of 1,206 and the true heat drop is 122, so the initial steam must have a heat content of $1{,}206 + 122 = 1{,}328$.

And an entropy of 1·587.

The steam table gives as the initial steam quality 475 psi.g. at 650° F.

This is confirmed by the Mollier chart.

Now in a sugar refinery there is always a risk that the process condensate may be contaminated with sugar. At high temperatures sugar is broken down into organic acids so that sweet water cannot be tolerated in a high pressure boiler. There are alternatives. Either the back pressure can be raised and all the exhaust can be passed into a still which will produce the process steam and the whole of the original steam will return as pure uncontaminated condensate in a closed circuit; or 100 per cent. treated make-up water can be used. A pressure of 475 psi may be considered rather high for the use of 100 per cent. treated London or other hard water so other means should be investigated.

146. HIGH PRESSURE OR HIGH SUPERHEAT. It is possible to get the same power from a lower pressure steam if much more superheat is used.

An inspection of the Mollier Chart shows that the pressure lines diverge in such a way that a given heat drop can be obtained over a smaller pressure drop, the higher the temperature. It is sometimes suggested that the disadvantages of high pressures are more serious than those of high temperatures, although this view is not shared by many people. Let us see therefore what can be done if the initial pressure be limited to 350 psi.a. with the other conditions as in Section 145. Steam at this pressure occupies a much larger volume and as we propose to use a considerable superheat, the volume will be greater still. We are therefore probably justified in taking the efficiency ratio at 71 per cent. With a net heat drop of 122, the 350 psi adiabatic heat drop will be 172 Btu.

On the Mollier chart find, by trial and error with a pair of dividers, the entropy co-ordinate such that the distance along it between the 350 psi.a. line and the 75 psi.a. line is 172 Btu.

This entropy is found to be 1·693.

And the state of steam at 350 psi.a. where its pressure line cuts the 1·693 entropy line gives it a temperature of 782° F.

Or a superheat of 350° F., its total heat being 1,409 Btu..

The exhaust at 75 psi.a. will have a total heat of 1,409 − 122 = 1,287 Btu.

The total heat of exhaust steam was stipulated in Section 145 to be 1,206 so that there will be 81 Btu of excess heat to be got rid of by desuperheating.

The latent heat at 75 psi.a. is 905.

So that for every pound of hot exhaust desuperheated there will be an evaporation of desuperheating water of $\frac{81}{905} = \cdot 0895$ lb.

The total amount of process steam per kWh will be.

$$28 \times 1 \cdot 0895 = 30 \cdot 5 \text{ lb./kWh.}$$

So this is not a solution. It will be necessary either to raise the superheat still further and work with a larger heat drop, or a dry desuperheater must be used. A few trial calculations will show that a really material increase in the heat drop will be necessary and that the steam will become so hot as to be probably unsuitable for operating a turbine outside a high-grade power station. It might be possible to desuperheat the exhaust steam with the boiler feed water, on its way from economiser to boiler, by passing it through a heat exchanger.

147. BACK PRESSURE AUXILIARIES. There is one important point that must always be borne in mind when designing a high pressure, back pressure plant. The object of going to high pressure is to generate more power. High pressure boilers call for high powered auxiliaries—the same kind of auxiliaries that the power station needs. Now the power these auxiliaries take depends on the output of steam, not the output of power. The auxiliaries of a power station working at about 650 psi will take 4 to 5 per cent. of the current generated, and the steam rate of the turbine will be about 9 lb./kWh. A back pressure plant working at 650 psi will require nearly the same auxiliary power per lb. of steam, but the turbine steam consumption may be 27 lb./kWh or three times that of the power station. The auxiliaries may therefore absorb 12 to 15 per cent. of the power generated.

When designing back pressure plants the turbo-generator alone must never be considered. The net power output after supplying the higher pressure boiler's extra needs is the only criterion. It may often pay to use a less elaborate and a lower pressure boiler plant, if by so doing the auxiliary load is largely lightened.

148. CHOICE OF MACHINE TYPE. There are all kinds of ways by which power can be generated in steam-using factories. The choice of method depends on many things. A pass-out turbine can only work really efficiently if the electrical load is fairly steady and if the pass-out quantity is fairly constant.

Any variation in electrical load alters the "natural" pressure at the pass-out point, and devices, such as adjustable diaphragms, have to be used to keep the pass-out pressure reasonably steady.

Again, any variation in the quantity of steam passed out changes the pressure drops on either side of the pass-out point. In Section 131 a particular machine was discussed and it was shown that a pass-out back-pressure machine could be expected to give 5,500 H.P. against 4,320 H.P. for the straight back-pressure set. These figures are definitely rose-tinted. They apply to specification load. They may only occur one day a month. Now the straight back-pressure set is stolid and simple and will operate with less loss of efficiency over a wider variation in conditions. The pass-out is more temperamental, is susceptible to altered conditions and must contain more complex control and governing gear. The inefficiency of a mixed pressure machine is even more marked under fluctuating conditions. While there are many admirable applications for pass-out and mixed pressure machines it is suggested that unless really substantial economies are to be expected, straight back-pressure and separate exhaust turbines are to be preferred.

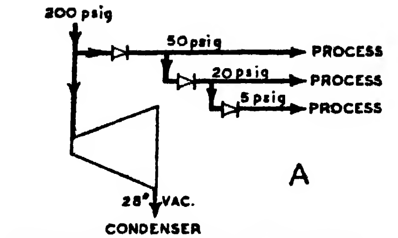
In small factories a turbine is probably unsuitable owing to its very low efficiency in small sizes. The small works must use reciprocators. In many processes the steam must be free from oil, for example, where the steam is to be injected into a food or a textile product. Oil separators will not take "more than somewhat" out of steam. Owing to the difficulty in removing oil, it is common in food and textile works to run back pressure reciprocators with saturated steam so that the cylinder condensate acts as lubricant and little or no oil is required. Consequently the exhaust will always be wet. We just have to make the best of this and do our utmost to remove the moisture before it reaches the process plant.

Fig. 42 shows some of the possible arrangements.

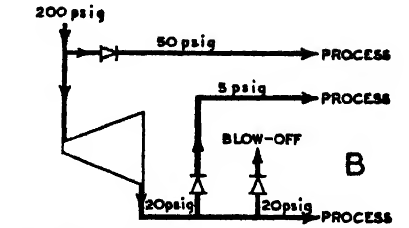
Apart from A which contains the maximum of thermodynamical wickedness, it is impossible to say which of the various schemes is the best without going into the details of each case. But D and G are obviously targets as they contain no reducing valves. Much depends on the fluctuations of both power and process demands, the proportion of the demands, the extent of the power shortage and the size of the installation. It is hoped that the foregoing may have helped to show at least that there may be more than one solution to any particular problem.

One important aspect must not be overlooked. Heat demands in some factories vary a great deal over the year; sometimes because the space heating is affected by winter or summer weather; sometimes because of seasonal variations in trade. There is no use making a nice plant which works well in winter but blows off steam in spring, summer and autumn. It may, however, pay to have a system where steam must be blown off for three or four months. Every case must be looked into on its merits.

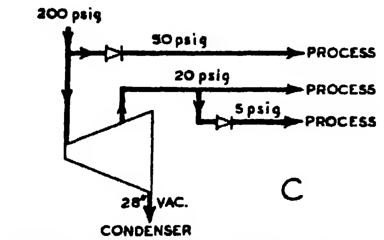
149. INDUSTRIAL POWER GENERATION. Let us take certain states of steam in common factory use and see just what such steam could do in power production under various conditions. We will take two factories, one ten times the size of the other. Some very interesting and important things will emerge.



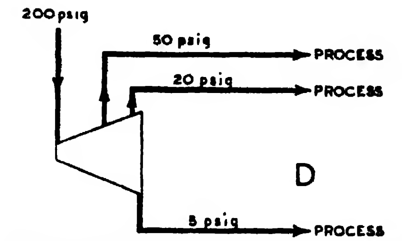
REDUCING VALVES & CONDENSING TURBINE



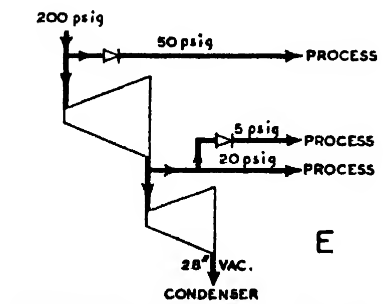
REDUCING VALVES, BACK-PRESSURE TURBINE & BLOW-OFF



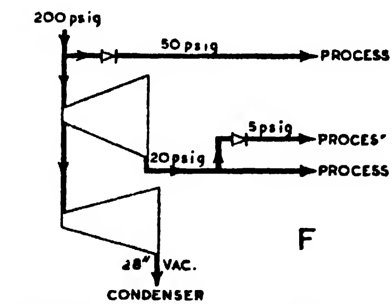
REDUCING VALVES & PASS-OUT CONDENSING TURBINE



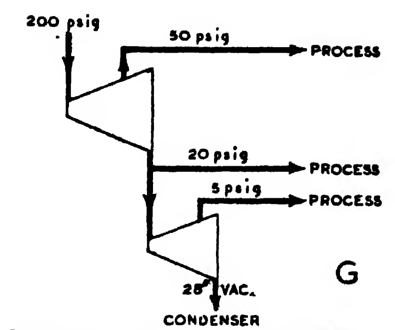
DOUBLE PASS-OUT BACK-PRESSURE TURBINE



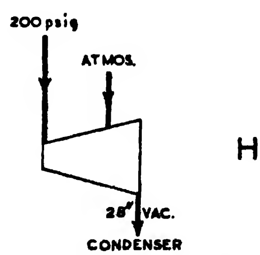
REDUCING VALVES, BACK PRESSURE & EXHAUST TURBINES



REDUCING VALVES, BACK PRESSURE & CONDENSING TURBINES



PASS-OUT BACK-PRESSURE & PASS-OUT EXHAUST TURBINES



MIXED PRESSURE TURBINE

FIG. 42. VARIOUS WAYS OF GENERATING POWER AND SUPPLYING PROCESS STEAM AT THREE DIFFERENT PRESSURES

First consider a factory using 12,500 lb. of steam per hour. By using the methods already described we get the results shown in Fig. 43. The cone slices are meant to represent engines or turbines working between the steam conditions shown. Certain conventions have been followed. Where the exhaust is at or above atmospheric pressure, conditions have been arranged to give dry saturated exhaust, because such exhaust could be used for process or in another machine. Where the inlet steam is at or below atmospheric pressure the inlet steam is taken as being dry saturated and the exhaust has been allowed to get wet. The figures have been rounded off and are only approximate.

Fig. 43 represents what a factory using 12,500 lb. of steam/hour could do. We will assume that it is raising steam for heating only which it uses at about 40 psi.g. If it raised its steam at 200 psi.g. with moderate superheat and passed it first through an engine A, Fig. 43a, it could generate 270 kW for about ·5 lb. coal/kWh. Suppose on the other hand that it was using 12,500 lb. of steam in a non-condensing engine exhausting to atmosphere A.B, this exhaust could give 270 kW if passed through an exhaust turbine C, connected to a condenser at the very modest vacuum of 24 in. If the vacuum were improved to 28¾ in. the exhaust turbine C.D. could generate 540 kW. All this exhaust-produced power has used no coal at all.

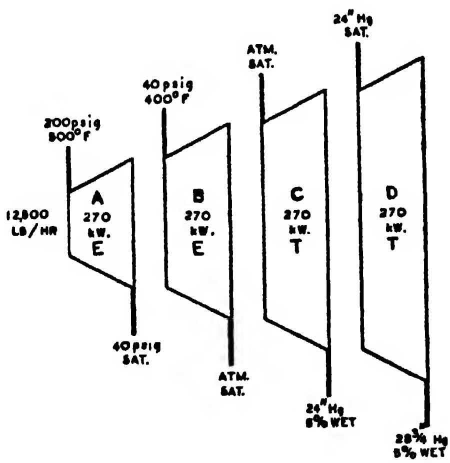


FIG. 43. INDUSTRIAL POWER GENERATION IN VARIOUS WAYS IN A SMALL FACTORY

Now look at Fig. 44 which shows what a factory might do using 10 times the steam. Any machines used here will be much larger and more efficient. It will be seen that each machine will give 3,000 kW for each of the chosen pressure drops. Now suppose the plant is working turbine B, that is to say it is raising steam at 200 psi and exhausting to process at 40 psi.g. and suppose that the process is primarily evaporation. The vapour is probably being condensed at about 24 in. vacuum. This factory could instal new boilers and could put in a vacuum turbine E. The new boilers would raise steam at 750 psi and the steam would pass through turbines A and B exhausting to process at 40 psi.

The heat would re-appear as vapour at 24 in. and the vapour would pass through turbine E finishing up in a condenser at 28¾ in. vacuum. The factory would thus generate an extra 6,000 kW. The 3,000 kW produced in turbine A would use ·4 lb. coal/kWh and the 3,000 kW produced by turbine E would use 0·0 lb. coal per kWh. The whole extra 6,000 kW would be generated for an average coal of ·2 lb./kWh.

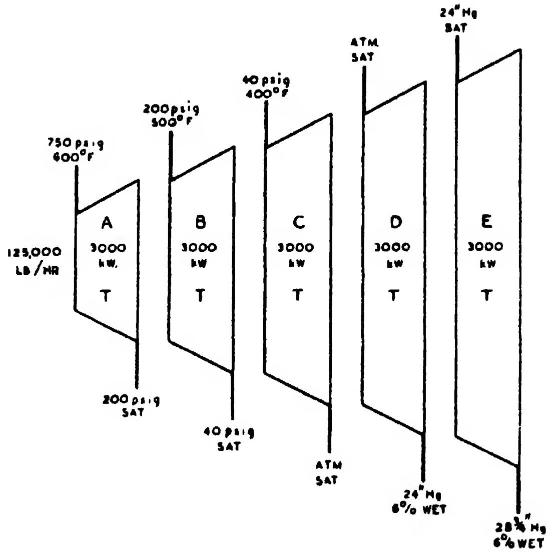


FIG. 44. INDUSTRIAL POWER GENERATION IN VARIOUS WAYS IN A LARGE FACTORY

Now suppose Fig. 44 represents a steel works raising steam at 200 psi for its rolling mill engines and its hammers. These are equivalent to machines B and C and total 8,000 H.P. By adding machines D and E and probably an exhaust accumulator, it could produce an extra 6,000 kW for an average consumption of coal of 0 lb./kWh. because D and E use no extra coal. The logical development of this is to use some of the surplus power to drive some of the rolling mills. Less steam will be needed and made. Less coal will be burnt. Less surplus power will be made. A nice balance must be struck.

150. VERY SMALL POWER USERS. Back pressure engines or turbines can often be used with great economy where steam is used, possibly in large quantities, but where the power load is so small that a power generating plant is not justified.

In some cases steam is used to heat calorifiers through which water is circulated. It is sometimes possible to use a little back pressure turbine to drive the circulating pump, the turbine exhaust blowing into the calorifier. Some quite successful little turbine-driven centrifugal pumps are working as slow as 1,250 r.p.m. Again, plenum ventilation heating systems often use steam for heating, while the fan is almost always electrically driven. Here is another good opening for the turbine, which would drive the fan and exhaust into the air heater.

Little turbines use a vast amount of steam per H.P., possibly several hundred lb./H.P./hr. This sometimes chokes off the potential user. *Provided the exhaust steam is all properly used, it does not matter how much steam is used.* A little turbine of 2 H.P. will use no more *heat* if it uses 500 lb. steam per hour than if it uses 250 lb.

Nowadays, many people seem to find difficulty in remembering that machines, pumps, fans, etc., do not necessarily have to be driven by electric motors. The steam drive is often superior. Speed can be varied at little or no extra cost, whereas electrical speed variation is very costly.

In cases where the steam/power ratio is such that it is impossible to generate more than a half or a quarter of the power, the driving of one or two of the larger machines by steam engine or turbine, and the remaining plant by bought electricity may sometimes be an elegant way of doing things. For example, a small brewery may be unable to generate more than half its power. It therefore decides to buy all its power requirements. It might be very economical to use a steam driven refrigerator instead of driving the compressor by electric motor.

151. ADAPTATION OF EXISTING MACHINES. From a consideration of the principles discussed in the foregoing, each factory must try to decide what the best arrangement would be with its particular conditions. Only in a few cases will an entirely new power plant be contemplated. It remains, therefore, to see how existing plant and its operation can be modified to get the most economical possible results.

152. UTILISING ATMOSPHERIC EXHAUST. Steam at atmospheric pressure can be used for most space heating purposes (except some high pressure and unit heater systems—the cheap capital cost systems). If the space heating is being done by low pressure water all that is required is a calorifier or a spray condenser (see Chapter 14).

If space heating is done by low pressure steam it may be possible that steam at 1 or 2 psi.g. will be adequate. Increasing the back pressure from 0 to 3 or 4 psi.g. will only affect a very fully- or over-loaded engine, but it may be necessary to lengthen the admission period in a throttle governed engine.

Every effort should be made to use the lowest possible pressure in the heating or process system so as to make the use of the engine exhaust as easy as possible.

If the engine is lightly loaded, or if it can be lightly loaded by running a spare engine as well, it may be possible to turn an engine exhausting to atmosphere into a back pressure engine.

153. CONDENSING ENGINES. It is very doubtful whether engines exhausting to condensers can be turned into back pressure machines, even if the back pressure is only 1 or 2 psi above atmosphere. It might be possible if the engine is only half or quarter loaded. It might also be possible to use the condenser cooling water for heating. It certainly should be if the vacuum is only about 20 in. (In the author's factory all space heating, air heating and

social hot water heating is done by means of condenser water from condensers working at 24 in. vacuum.) If there is a spare engine it might be converted to back pressure working and the balance of the power requirements made up from the condensing engine, but the no-load steam consumption and the Willans Line (Sections 121 and 122) must be considered.

154. CONDENSING TURBINES. The same remarks apply to condensing turbines as to condensing engines, only more so. A turbine is more susceptible to exhaust pressure variations than a reciprocator. Any raising of the back pressure in a turbine will reduce the power produced proportionally rather more than in the corresponding reciprocator.

155. CONVERTING NON-CONDENSING ENGINES TO CONDENSING. No such difficulties apply to the conversion of engines exhausting to atmosphere into condensing engines. Where it is impossible to find a use for the exhaust steam, an economy can almost always be made by exhausting the engine into a condenser. The most suitable engines are those that run continuously. Intermittent engines, winders and rolling mills, do not show such economies because much water has to be pumped through the condenser when the engine is not working. When an engine is converted to condensing from non-condensing, it is important that indicator diagrams be taken and the valve gear adjusted to suit the new conditions (see Chapter 7). A very high vacuum should not be aimed at. It calls for much water, large vacuum pumps and cannot be taken advantage of by an engine that is not designed as a condensing engine. 18 in. or 20 in. is probably adequate.

156. CONVERTING EXISTING MACHINES TO PASS-OUT WORKING. As regards turbines this is not a very hopeful proposition. There is probably no connection in a suitable position on the turbine casing. Even if there is, it is by no means certain that satisfactory results will be obtained, although such a conversion was fairly successfully made on the turbine in the author's factory. It has been explained in Sections 130 and 148 that for correct pass-out working the pressure drops and therefore blade areas, before and after the pass-out point, must be very carefully proportioned. To pass steam out of an existing machine may not be satisfactory without fairly major alterations and the inclusion of pretty elaborate controls, unless the pass-out quantity is small, and/or the pass-out pressure can be allowed to fluctuate.

Compound reciprocators are quite a different matter and offer much more hopeful opportunities. With a tandem compound engine very large amounts of steam can be passed out of the steam receiver between high and low pressure cylinders. The limit is set by the amount of work that the high-pressure cylinders can do on full admission, and by the point at which the low-pressure cylinder ceases to be a power producer or even a passenger and becomes a pump. Much can be done by hanky-panky with the valve gear.

Let us take an imaginary factory. 20,000 lb. of steam an hour is raised at 150 psi.g. Half, 10,000 lb./hour, goes to a tandem compound condensing engine. The remaining 10,000 lb./hour goes to process at 20 psi.g. through a reducing valve.

Assume the engine is working at about 33 per cent. cut-off in the high-pressure cylinder, that the power is developed in equal parts by high-pressure and low-pressure cylinders and that the receiver pressure is about 22 psi.g. the exhaust being at 20-in. vacuum. Let us say that we will draw all the process steam from the receiver. This means that 10,000 lb. of process steam/hour will go through the high-pressure cylinder and will consequently develop half the load. The remaining half load must be produced by steam that goes through both cylinders and exhausts to the condenser. If the valve gear can be so adjusted that this steam does half work in the high-pressure and half in the low-pressure cylinder we shall only need to pass 5,000 lb./hour to the condenser.

By putting all the steam into the engine and passing out the process steam after the high pressure cylinder the total steam consumption will have been reduced from 20,000 lb./hour to 15,000 lb./hour. This is equivalent to a saving of about 700 tons of coal a year on one shift, or 2,000 tons a year on three shifts.

An alteration such as this may not be possible with a cross compound engine. In the example given above, three-quarters of the load is taken by the high-pressure cylinder, and one-quarter by the low-pressure cylinder. This might well cause rough running and bearing trouble in a cross compound, but it is entirely unobjectionable in a tandem.

157. MAKING FULLEST USE OF BACK PRESSURE PLANTS. Every back pressure machine in the country, for whose exhaust a REAL economic use can be found, should be working to full load. In 1942 there was a large modern factory equipped with back pressure turbines. It was using all its process steam at 10 to 15 psi blown down from boiler pressure through reducing valves ; it was keeping its back pressure plant as spare, and taking all its power from the grid, except for its largest power user which was turbine driven exhausting into a condenser ! The reason was that the chief engineer was costing his power by multiplying the adiabatic heat drop by the steam consumption. This may well have given him a steam cost of nearly twice the correct figure. This has already been discussed in Section 120 and is discussed again in Chapter 21.

Let us say it again : *Every back pressure machine, for whose exhaust there is a real use, should be working at full load.* If there is no real economic use for either the exhaust steam or the power in the producing factory, can exhaust steam be supplied to neighbours or power supplied to the supply company ?

The important thing to remember is that a steam engine is a heat engine which is inherently very wasteful. We have not yet learnt how to extract more than about one-third of the energy as power, but if we remember that the machines are HEAT engines and look after the HEAT we can turn them into machines with efficiencies of 60, 70, 80 per cent.

158. INTERNAL COMBUSTION ENGINES. Combined power and heating systems are common in steam plants, though not so common or so thorough-going as they ought to be. Combined power and heating systems with internal combustion plants are rare, although it is exceedingly simple to carry out an elementary combination. (Although this subject is hardly steam technology, no excuse is needed for discussing it.)

Every internal combustion engine has to have some way of cooling the cylinder. Small air-cooled engines might have the cooling air ducted into a room that requires warming. Water-cooled engines have a radiator or cooling tank put in the coldest place the better to waste the heat. Why not pipe this water round the radiators of the office or store or workshop ? If the load fluctuates, it may be necessary to have a big tank to store the heat—but put the tank indoors.

It is important that the water passed through the jackets should be kept in a closed circuit, if the cooling water is hard water, lest scale be deposited inside the jackets and cause the cylinders to overheat.

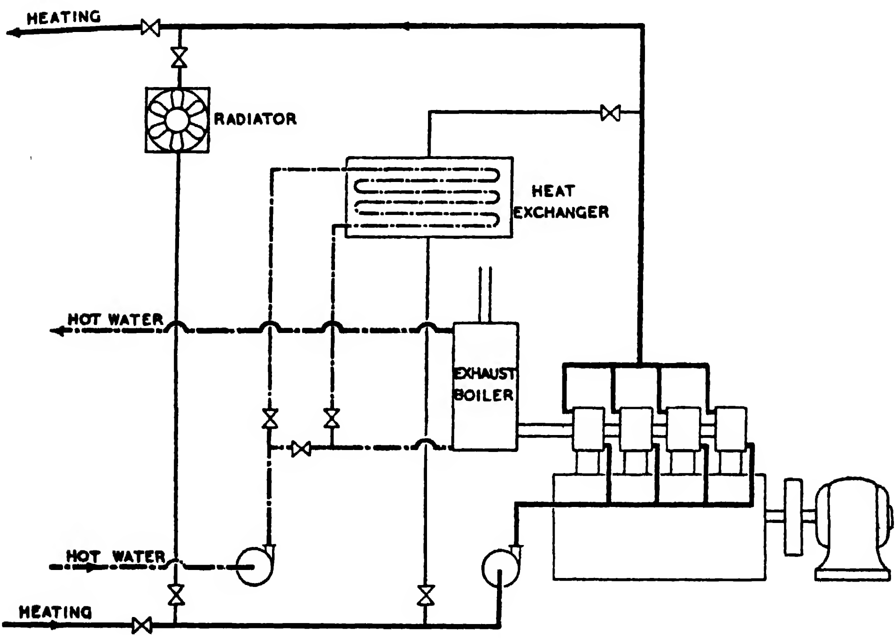


FIG. 45. COMBINED HEATING, HOT WATER AND POWER PLANT WITH INTERNAL COMBUSTION ENGINE

When jacket water is used for space or process water heating it should preferably be thermostatically controlled because the engine will be inefficient if the jackets are too cool. Provided the water does not boil, it generally does not matter if the water is on the hot side. The engine makers should be consulted as to the best working jacket water temperature.

The exhaust of internal combustion engines contains a lot of heat, not as much as steam engine exhaust, and it is dirty and not so usable ; but 30 per cent. of the fuel heat is thrown away in the exhaust. Exhaust boilers are made and can be very successful. Such boilers can either heat water or raise steam.

159. EFFICIENCY OF COMBINED INTERNAL COMBUSTION POWER AND HEATING. A small 20-kW diesel engine installation is running in Kent and giving an overall efficiency of about 67 per cent. The

jacket water is connected to the space heating circuit while a thimble tube exhaust boiler is connected to the hot water circuit. The heat distribution is approximately thus :—

Power	20 per cent.
Jackets to heating	25 „ „
Exhaust to hot water	22 „ „
	67 „ „

A heat exchanger is fitted between the two circuits to enable all the engine heat to go into one circuit when required. A radiator is fitted in the jacket circuit for occasional use when no heat is needed. The plant lay-out is shown in Fig. 45.

A much larger installation (four 300-kW sets) on exactly similar lines is running at the Bank of England. The overall efficiency of this plant is about 75 per cent.

* * *

This Chapter has not been written to encourage a factory staff to design or radically modify its own power plant. No one without great experience can foretell accurately the performance of back pressure, pass-out and exhaust machines. It is hoped, however, that the investigation of the problems involved will enable factory management to put before the power plant builders schemes which are not fantastic and will enable them to discuss such schemes intelligently with the indispensable experts—and—to keep their back pressure plants on full load.

* * *

CHAPTER 4

PREVENTING THE ESCAPE OF HEAT—LAGGING

With the skin he made him mittens,
Made them with the fur side inside,
Made them with the skin side outside.
He, to get the warm side inside,
Put the inside skin side outside;
He, to get the cold side outside
Put the warm side fur side inside.

G. A. STRONG. *The Song of Milkanwatha.* 1912 ?

STEAM is simply a vehicle for conveying heat. In order to bring heat to the right place, we must, as far as possible, prevent heat escaping to places where it is not wanted. Apart from steam leaks, which are so obviously wrong as hardly to need mention, steam loses heat from the pipes and vessels that contain it to the surrounding air. Of all the methods of saving heat, the curing of leaks and the prevention of heat leakage by "lagging" or covering the hot surface are the easiest and most straightforward.

160. LEAKS. Leaks are much more important than they look. If plant is allowed to leak steam, it generally means that the standard is low and that there will be many leaks, not only of steam, but of air, water and other things. Suppose a valve spindle is badly packed so that there is a space of a hundredth of an inch between the spindle and the packing. If the spindle is ¾-in. diameter. the area of the leakage ring will be nearly the equivalent of a 3/32-in. hole. Suppose there are 50 such leaks in the plant, these will add up to the equivalent of a hole ⅝-in. in diameter. Table XVI shows the amount of steam, water and air that will leak through small holes.

TABLE XVI. FLUID LOSS BY LEAKAGE

DIAMETER OF HOLE	STEAM—LB./HOUR		WATER—GALLS./HOUR		CU. FT. FREE AIR/MIN. 80 PSI
	100 PSI	300 PSI	20 PSI	100 PSI	
1/16″	14	33	20	45	4
⅛″	56	132	80	180	16
3/16″	126	297	180	405	36
¼″	224	528	320	720	64

Table XVII shows the tonnage of coal lost per year due to the steam leaks given in Table XVI. The really staggering size of these losses is seldom realised. Is there a factory in the country that can truly say that its leaks do not add up to a hole ⅛ in. dia. with a consequent loss of 10/50 tons of coal a year ?

One of the principal causes of leaks is cheap pipe flanges with too few bolts. Table XVIII on pages 847–849 gives a list of British Standard Pipe Flanges for

various pressures. In the author's factory the modern pipes erected to these standards can almost be said to be leak-proof, whereas the old piping with fewer bolts leaks profusely.

TABLE XVII. COAL LOSS BY STEAM LEAKAGE

DIAMETER OF HOLE	TONS OF COAL PER ANNUM			
	100 PSI BOILER EFFICIENCY 65%		300 PSI BOILER EFFICIENCY 75%	
	48 HR. WEEK	144 HR. WEEK	48 HR. WEEK	144 HR. WEEK
$\frac{1}{16}$"	2	7	5	14
$\frac{1}{8}$"	9	28	19	56
$\frac{3}{16}$"	21	62	42	126
$\frac{1}{4}$"	37	111	74	223

161. CONDUCTION, CONVECTION AND RADIATION LOSSES. In Section 18 the mechanism of heat transfer was briefly discussed. A hot surface, pipe, tank, pan, etc., parts with its heat to the air by convection and to all the surroundings by radiation. At low temperatures the loss by true radiation forms only a small proportion of the whole. As the temperature rises radiation plays an increasingly important part. The amount of heat radiated by a body is proportional to the fourth power of the absolute temperature. For example :—

Temperature ° F.	*Relative Heat Loss by True Radiation only*
100	1·0
200	1·9
300	3·4
400	5·6
500	8·6
600	12·8
700	18·4
800	25·6
900	34·8
1,000	46·2

As the temperature rises convection also is more rapid, so that there are two causes tending to make the heat loss increase in more than direct proportion to the temperature.

The object of lagging is two-fold. It is to reduce the temperature of the exposed surface so that heat will be lost less rapidly by convection. It is to interpose an opaque layer between the hot pipe and the surroundings so as to prevent heat loss by radiation.

162. PROPERTIES OF AN IDEAL LAGGING MATERIAL. As has been explained in Section 18 materials vary greatly in the ease with which they conduct heat. Silver and copper conduct heat very easily—they have a high

thermal conductivity. Air conducts heat with great reluctance. Air is about 10,000 times as bad a conductor as copper. So air would seem to be a good material to use for preventing the escape of heat ; in fact as our pipes are already encased in air what could be better ? Unfortunately, air lets all the radiation through, and air currents form with the greatest ease with a very small local change of temperature. Although air is such a bad conductor, immediately a film of air does get warmed it rises at once and is replaced by cold air. Short distance conduction, current formation and heated film replacement are the mechanism of convection and result in a very quick and large loss of heat. To prevent convection means preventing air currents. To prevent radiation means interposing some substance that is opaque to heat radiations. Now if the material that is interposed to intercept the radiations does not reflect them back, it must absorb them and will itself get rapidly heated by radiation ; so it should be a very bad conductor so as not to act as a re-radiator. Very few solids are as bad conductors as gases. The best material to use is one that consists of a microscopic sponge, all the cells being filled with air, and the material should be a bad conductor and should be opaque or should reflect. The air cells should be very small so that convection currents cannot form inside them. The material should be mechanically robust and should not deteriorate with heat. Table XIX gives the conductivity of a number of different materials in Btu conducted per sq. ft. per hour per ° F. temperature difference between either side of one inch thickness of the material at low to moderate temperatures.

TABLE XIX. HEAT CONDUCTIVITY

MATERIAL		APPROXIMATE BTU/SQ. FT./HOUR/ °F. DIFF./1" THICKNESS
Heating Surfaces	Copper	2,620
	Aluminium	1,430
	Brass	720
	Cast iron	340
	Steel	310
	Lead	240
Resistant Films	Water at 32° F.	4
	Water at 200° F.	5
	Air	·2
	Scale	1 to 12
Lagging Materials	Diatomite, Kieselguhr or fossil meal — Poor	1·0
	Diatomite, Kieselguhr or fossil meal — Good	·6
	85% magnesia	·4
	Asbestos flock	·4
	Glass wool	·4
	Animal hair	·3
	Cork	·3

All kinds of conductivity figures are given by different authorities and by different vendors of lagging materials. Table XIX is an average of a number of figures given by different authorities and brought to round numbers. The

good insulating properties of the lagging materials are largely due to their structure which entangles the maximum of air within a non-conducting opaque sponge or felt.

163. CHOICE OF LAGGING MATERIAL. Cork is probably the best of all materials, due to its intricate natural structure. It is however inflammable and gets charred and brittle at moderate temperature. It is therefore seldom used on steam plant, but it is in almost universal use on all high class refrigerators. Animal hair is excellent, but it is inflammable and can harbour livestock. Asbestos and 85 per cent. magnesia are really the outstanding materials for industrial lagging, although glass-wool is becoming increasingly used, and is said to have the advantage that it does not so easily become water-logged. Asbestos is a natural mineral consisting of a variety of complicated silicate compounds generally containing calcium or iron or both. It is built up of fine compressed fibres which can be teased out into a first rate fire-proof air-entangling felt of very low conductivity. 85 per cent. magnesia consists of a mixture of 85 per cent. magnesium carbonate with 15 per cent. asbestos fibre to bind the fine magnesia crystals together. Conflicting claims are made for each of these materials. Actually there is little to choose between them—both are excellent. Magnesia should, however, not be used on a surface hotter than about 550° F. to 600° F. Other materials are kieselguhr or diatomite and various degrades of adulterated asbestos and magnesia. There is little money saving in using these poorer quality materials, but there is a definite loss in heat saving. As far as this book is concerned, " lagging " means 85 per cent. magnesia, glass wool or first class asbestos.

164. APPLYING THE LAGGING. Lagging can be obtained in bulk form to be applied plaster fashion, or it can be bought in moulded sections—half cylinders for pipes, half boxes for flanges, slabs for tanks, etc. It generally pays to use the moulded forms. Lagging is fragile and is easily damaged. When repairing damage it is easy to take off a moulded section and replace it. Much of the damaged piece can be cut up into wedge sections for covering bends and awkward shapes. When a pipe is taken down most of the plastered lagging is generally lost, whereas the sections can be recovered intact with a little care. Moulded sections can be fitted during the erection of the plant on cold pipes and tanks. Plastic lagging must be applied to hot surfaces. Moulded lagging can be applied much more quickly than plastic lagging and very little skill is needed.

Lagging should almost always be protected by a covering. All lagging materials are porous and must be protected from damp, otherwise the air spaces would get water-logged and the insulating qualities would dwindle. Lagging is very fragile and easily damaged. In the author's factory steel-sheeted lagging lasts three to five times as long as unprotected lagging and only costs about 30 per cent. more. Fig. 46 shows the method of fastening the sections to the pipe and securing sheet steel covers. The moulded sections are attached to the pipe by a spiral binding of soft wire—about 19 S.W.G. For a really neat job the sheeting should be secured by means of special sheet metal screws—Parker Kalon screws. A quicker rougher job can be made with soft wire

binding. The wire should be twisted up fairly tight and when all the wires are in position on one section the wire can be given a quarter kink with a pair of pliers to do the final tightening without risk of snapping the wire at the twist. The sheeting should be about 22 S.W.G. If it is galvanised it need not be painted on the outside. If it is black it should be given a coat of lead paint and a final coating of aluminium paint. Whether galvanised or black it should be well tarred on the inside. The sheet should be passed through bending rolls to give it the correct trough shape.

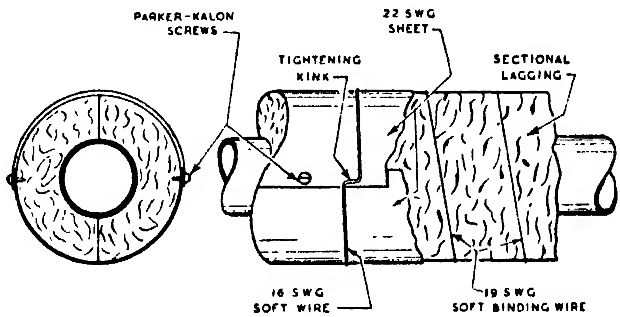


FIG. 46. SECTIONAL MOULDED LAGGING PROTECTED BY SHEET STEEL

165. LAGGING THICKNESS. The most important thing about lagging is that there should be SOME lagging. It is far more important that everything hot should be lagged than that some plant should be beautifully covered and flanges and oddments left bare.

On a flat surface the thicker the lagging the better (within economic reason), but on a small diameter pipe things are different. The thicker the lagging put on a pipe the larger is the final heat-losing surface. Table XX shows the temperature of the outer surface of the lagging on a 6 in. pipe with various lagging thicknesses at various temperatures. It is obvious that with low temperature pipes the gain by going to thick lagging cannot bring a proper return except in very special cases. Some general guidance can, however, be given.

The smaller the pipe diameter the thinner should be the lagging. If the pipe is only ½ in. bore, its surface with 1 in. of lagging will have a diameter of 2½ in., whereas with 3 in. of lagging the surface diameter would be 6½ in. Heat loss is proportional (over a limited range) to the temperature drop. Suppose our ½ in. pipe had a temperature of 250° F. and that its outside surface temperatures were the same as in Table XX, the pipe with 1 in. lagging would have a temperature drop over the surrounding air of 29° F., while the pipe with 3 in. lagging would have a temperature drop of 12° F. The relative heat losses would be about 72·5 with 1 in. and about 78 with 3 in. The thick lagging, apart from costing about three times as much gives if anything more heat loss. This is only a rough general statement. With cheap bad lagging, a thick covering on a small pipe gives a bigger heat loss than a thin covering, but good quality lagging is said always to reduce the total heat loss with increased

thickness, though the gain is quite uneconomic. A rough rule is that lagging need not usually be thicker than the pipe diameter, and that the smaller the pipe the more important is it to use the best quality lagging.

An inspection of Table XX shows that lagging is an excellent example of the law of diminishing returns. It is the first thickness of lagging that really matters.

TABLE XX. APPROXIMATE OUTSIDE TEMPERATURES OF LAGGED SURFACES—6 IN. BORE STEEL PIPE, STILL AIR AT 70° F.

INTERNAL TEMPERATURE °F.	TEMPERATURE OF LAGGING °F.					
	LAGGING THICKNESS					
	1"	1½"	2"	2½"	3"	4"
100	75	74	73	72	72	71
150	83	80	78	76	75	74
200	91	86	83	80	79	76
250	99	92	88	84	82	79
300	107	98	93	88	85	82
350	115	105	98	92	88	84
400	123	111	102	96	91	87
450	131	117	107	100	95	89
500	139	123	112	104	98	92
600	155	135	122	112	104	98
700	171	147	132	120	110	102
800	188	160	142	128	117	107

TABLE XXI. RECOMMENDED LAGGING THICKNESSES

INTERNAL TEMPERATURE °F.	THICKNESS OF LAGGING				
	HIGH TEMPERATURE COMPOUND	85% MAGNESIA OR ASBESTOS			
		PIPE BELOW 3" DIA.	PIPE 3" TO 6" DIA.	PIPE 6" TO 9" DIA.	PIPE OVER 9" DIA. AND FLAT
Up to 200	—	1"	1"	1"	1"
200–300	—	1"	1"	1½"	2"
300–400	—	1"	1½"	2"	2½"
400–500	—	1½"	2"	2"	2½"
500–600	—	1½"	2"	2½"	3"
600–700	½" plus	2"	2"	2½"	3"
700–800	1" plus	2"	2"	2½"	3"

Table XXI gives suggestions for lagging thickness on pipes of various diameters for various temperatures. Above 600° F. some types of good quality lagging deteriorate and a special high-temperature-resisting compound should be applied in a thin layer next to the pipe.

Such things as hot water storage tanks might well have a little extra. The lagging of a steam accumulator is particularly important and should probably be 4 to 6 inches thick.

166. SAVINGS BY AND COSTS OF LAGGING. The losses from a hot surface are expressed in Btu per sq. ft. per hour. By working out the area of the hot surface the heat saving to be secured by lagging can be found from Tables XXII and XXIII. Fig. 47 gives the same information as Table XXIII and enables intermediate values to be obtained without interpolation. Having found the heat saving in Btu per hour, this must be worked back to coal as follows.

TABLE XXII. APPROXIMATE HEAT LOSSES FROM BARE AND LAGGED FLAT SURFACES IN STILL AIR AT 70° F.

INTERNAL TEMPERATURE °F.	BTU/SQ. FT./HOUR						
	LAGGING THICKNESS						
	BARE	1″	1½″	2″	2½″	3″	4″
100	60	10	7	—	—	—	—
150	155	25	20	—	—	—	—
200	295	45	30	25	—	—	—
250	450	65	45	35	—	—	—
300	645	85	60	45	40	—	—
350	875	105	75	60	45	—	—
400	1,140	125	90	70	55	45	—
450	1,450	150	105	80	65	55	—
500	1,810	175	125	95	75	65	—
600	2,670	—	160	120	100	85	65
700	3,750	—	—	150	125	105	80
800	5,100	—	—	185	150	130	100

Suppose we have a covered tank 9 ft. × 12 ft. × 6 ft. resting on the floor and containing hot water at 210° F.

The area of the four sides and top is

$$(6 \times 12)\,2 + (6 \times 9)\,2 + (12 \times 9) = 360 \text{ sq. ft.}$$

Interpolating in Table XXII gives the bare surface loss as 326 Btu/sq. ft./hr., and the loss with 1 in. lagging as 49 Btu.

Table XXI suggests that 1 in. lagging will be adequate for the temperature.

The saving by lagging is 277 Btu/sq. ft./hr.

The total saving is $277 \times 360 = 100{,}000$ Btu/hr.

If the boiler is 65 per cent. efficient and the coal has a calorific value of 12,000 Btu/lb., the coal saved is $\dfrac{100{,}000}{12{,}000 \times \cdot 65}$ 12·8 lb./hr.

TABLE XXIII. APPROXIMATE HEAT LOSSES FROM BARE AND LAGGED 6-IN. PIPE IN STILL AIR AT 70° F.

INTERNAL TEMPERATURE °F.	BTU/SQ. FT./HOUR						
	LAGGING THICKNESS						
	BARE	1"	1½"	2"	2½"	3"	4"
100	58	11	8·5	—	—	—	—
150	166	31	23	—	—	—	—
200	296	52	39	32	—	—	—
250	454	74	55	45	—	—	—
300	646	98	73	60	51	—	—
350	876	122	91	75	64	—	—
400	1,142	148	110	90	77	68	—
450	1,452	176	131	106	91	80	—
500	1,810	204	152	123	105	93	—
600	2,671	—	221	160	136	120	100
700	3,750	—	—	200	170	150	124
800	5,110	—	—	242	207	182	151

If the tank is hot for 50 hours a week, this totals 15 tons of coal a year. If the tank is hot for 150 hours a week, the coal saving is 44½ tons a year. So that without any other costs at all, the heat saving on coal alone due to the lagging is, if the coal costs 100*s.* a ton, £74 or £220 per annum depending on whether the factory is working one shift or three. The cost of lagging such a tank would be, in 1957, in the author's factory, labour and material only :—

	£	*s.*	*d.*	
Lagging—1-in. slabs	36	0	0	
Sheeting	13	4	7	
Labour	21	14	6	
	£70	19	1	say £75 all in.

Now let us take a 6-in. steam pipe carrying saturated steam at 120 psi.

The internal temperature will be 350° F.

Table XXIII shows the heat loss from the bare pipe to be 876 Btu/sq. ft./hr.

If the pipe is covered with 2 in. of lagging the loss is reduced to 75 Btu—a saving of 800 Btu/sq. ft./hr.

Let us assume that we have 100 ft. of piping and that flanges occur, due to valves, bends, etc., every 9 ft.

There will be 11 pairs of flanges and they will be good radiators and must be covered.

Table XXIV gives the area per foot run of various sizes of pipe and the area of a standard flange.

Eleven flanges will occupy about 2 ft. of the pipe length

so the area will be $(98 \times 1{\cdot}71) + (11 \times 1{\cdot}81) = 187{\cdot}5$ sq. ft.

If the boiler house is 70 per cent. efficient and the coal has a calorific value of 12,000 Btu/lb. and costs 100*s*. a ton, the saving per week of 50 hours is

$$\frac{800 \times 187{\cdot}5 \times 50 \times 100}{12{,}000 \times {\cdot}7 \times 2{,}240} = £1\ 19s.\ 10d.$$

This gives an annual saving on one shift of £100 and on three shifts £300.

The cost of lagging, 2-in. moulded sections, sheeting, paint and labour was (in 1957 in the author's factory) :—

	£	*s.*	*d.*	
98 ft. at 18*s*. 1*d*.	88	13	2	
11 flanges at 20*s*. 2½*d*.	11	2	6	
	£99	15	8	say £100.

Clearly, lagging pays and pays handsomely. The figures given are for really first-class steel-sheeted work.

Much lagging is protected by means of half an inch of hard setting compound reinforced with 1-in. galvanized wire netting. The outside surface having a covering of hessian or canvas trowelled in. In many factories this may prove quite satisfactory, but the author has found that the steel sheeting is more lasting and can be applied much quicker and requires less skill.

167. LAGGING FLANGES. In Table XXIV it will be seen that one flange is about equivalent to 1-ft. run of pipe. Lagging 1 ft. of 6-in. pipe costs 18*s*. 1*d*. and saves 18*s*. 9½*d*. a year. Lagging a flange costs £1 0*s*. 2½*d*. and saves 19*s*. 2*d*. So that, even on one shift, lagging flanges pays in less than 15 months. Many engineers refuse to lag flanges for fear that leaks may go undetected and that leakage may corrode the flange bolts. There are several good replies to this excuse. First, if moulded box lagging is used for flanges, a 3-in. length of ¼-in. pipe can be inserted into the bottom of the box to give early warning of any leak. Second, a really bad leak will soon make itself known. Third, lagging the flanges is one of the best ways of stopping leaks. Bare flanges on hot lagged pipes introduce temperature stresses at the flanges which may be the cause of the leak. In every ship's engine room the flanges are lagged as a matter of course. Were they left bare the engine room would be even hotter than it is. Ships' engine rooms do not suffer from leaking lagged flanges.

TABLE XXIV. AREAS OF PIPE SURFACES

BORE	SQ. FT. (STEEL PIPE UP TO 250 PSI) — PER FOOT LENGTH PARALLEL PIPE	SQ. FT. (STEEL PIPE UP TO 250 PSI) — FLANGE (2 EDGES 2 SIDES)
½"	·221	·331
¾"	·278	·327
1"	·352	·386
1¼"	·500	·488
2"	·622	·655
2½"	·785	·760
3"	·916	·931
4"	1·178	1·134
5"	1·440	1·638
6"	1·710	1·810
7"	1·963	2·169
8"	2·23	2·45
9"	2·49	2·98
10"	2·75	3·17
12"	3·27	3·85
15"	4·19	4·94
18"	4·98	6·22
24"	6·55	9·38

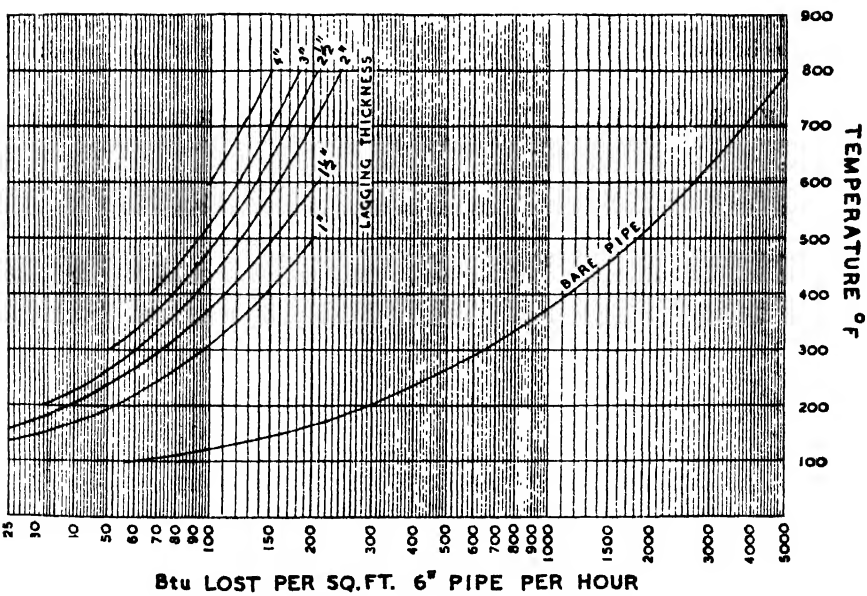


FIG. 47. HEAT LOSS FROM 6-IN. PIPE, BARE AND WITH LAGGING OF VARIOUS THICKNESSES

Special moulded flange boxes are probably not so satisfactory as a short length of moulded pipe lagging of a diameter to fit the flange. The flange covering should be sheeted and the edges of the flange lagging should be finished with a little plastic composition as shown in Fig. 48.

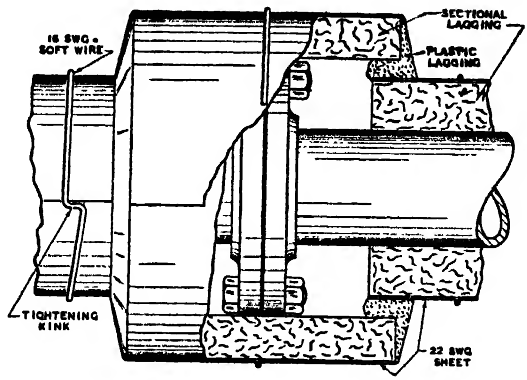


FIG. 48. SECTIONAL MOULDED LAGGING ON PIPE FLANGE PROTECTED BY SHEET STEEL

168. ASBESTOS ROPE. For very small pipes where only a thin lagging is justified, asbestos rope, though untidy, dirty and relatively short-lived, is useful. For temporary jobs it is often very convenient as it can be put on quickly and can be used over again. Table XXV gives the length of rope required.

TABLE XXV. LENGTH OF ASBESTOS ROPE NEEDED TO WRAP PIPES

PIPE BORE	LENGTH IN FEET PER FOOT OF PIPE				
	½" ROPE	¾" ROPE	1" ROPE	1¼" ROPE	2" ROPE
½"	11				
¾"	12	10			
1"	14	11	10		
1½"	17	14	12	10	
2"	20	16	14	11	10
3"	27	21	17	14	12
4"	34	25	20	16	14
6"	47	34	27	20	17

169. ALUMINIUM PAINT. Aluminium paint has been proved to save about 14 per cent. of whatever heat is being lost. A bare pipe at 250° F. loses 450 Btu/sq. ft./hr. By painting the pipe with aluminium paint the loss can be reduced to 387 Btu, a saving of 63 Btu/sq. ft./hr. This represents 3*s.* 2*d.* a

week on one shift on the 100-ft. 6-in. pipe that we have been considering in Section 166, or £8 3s. 8d. a year. The saving by painting a lagged pipe with aluminium paint is so small that it is not worth doing for the saving alone, but as the sheeting, or even plain lagging must be painted, it might as well be painted with aluminium paint, which looks very clean and nice apart from its other properties. The cost of aluminium paint compared with the cost of grey lead paint was :—

Cost of paint to cover 100 sq. ft. (paint only)—

	1944	1957
Grey lead—		
Priming coat	2*s*. 5*d*.	7*s*. 3*d*.
Finishing	2*s*. 8*d*.	7*s*. 4*d*.
Aluminium—		
Priming	3*s*. 1*d*.	6*s*. 8*d*.
Finishing	3*s*. 9*d*.	5*s*. 9*d*.

Aluminium paint should be a straight paint. A varnish base or a coating of varnish is said to destroy its saving grace. Aluminium paint should not be used on such high temperature plant as would turn it brown—not above, say 400° F.

Tests carried out in the author's factory in 1937 on various laggings and paintings on a length of 4-in. pipe carrying low-pressure steam gave :—

	Heat loss	
Bare pipe	100	
Bare pipe, aluminium painted	84	
		Saving 16 per cent.
Pipe lagged and sheeted unpainted ..	24	
Pipe lagged and sheeted, aluminium painted	21	
		Saving 12½ per cent.

TABLE XXVI. APPROXIMATE EFFECT OF AIR MOVEMENT ON HEAT LOSS

RELATIVE HEAT LOSS	AIR VELOCITY	
	FEET PER SECOND	MILES PER HOUR
1	Still	
1·5	4	3
2	8·5	6
2·5	13	9
3	18	12
3·5	24	16
4	30	21

170. AIR MOVEMENT. The heat losses given in Tables XXII and XXIII apply to still air. In practice air movement is always present. Table XXVI shows the effect of air movement on the rate of heat loss.

This Table shows us why our car boils on a hill as well as how serious it is to have unlagged plant near windows or doors. Air movement is obviously much more serious and important on bare pipes than on lagged pipes.

The heat loss from the liquid surface in an open tank is considered in Section 464, where it is shown that it is generally more important to cover the top than to lag the sides.

* * *

As most of the discussion in this chapter has been based on the published figures of heat losses done under test conditions with the lagging in first-rate condition, it may be thought that the figures given of the benefits of lagging are too rosy. This is far from being so. The savings given are probably understatements. Owing to the fact that in all factories draughts, from mild zephyrs to raging tornados, blow everywhere, any deterioration in lagging over test lagging is more than outweighed by the much greater heat loss that is actually taking place than the losses given in Tables XXII and XXIII. Tests done by the author actually inside the factory have given losses even in sheltered places considerably greater than those in the Tables.

* * *

CHAPTER 5

STEAM DISTRIBUTION—STEAM QUALITY

The sudden twist may stretch the swelling vein,
The cracking joint unhinge.

JOHN GAY. *Trivia.* 1716.

THE discussion of the two subjects, Distribution and Quality, in the same chapter may seem incongruous, but this is not so. The quality of steam is closely linked with its distribution. Heat losses during distribution have been dealt with in Chapter 4 and it is assumed in this chapter that all pipes are properly lagged. The quality of steam for power is plain sailing. The steam should be as hot and as high in pressure as circumstances allow. For nearly all processes steam should be at as low a pressure as possible and should not be highly superheated. How do these requirements affect distribution ?

171. THE FLOW OF STEAM THROUGH PIPES. The flow of steam, or any other fluid, depends on the pressure difference along the pipe, the resistance of the pipe to flow and the physical characteristics of the fluid—in this case steam. All kinds of complicated formulae are given to enable the flow to be calculated. Some of these involve such expressions as $P_1^{1\cdot9375}$. The author refuses to believe that the Almighty decided, when creating the world, that the flow of steam should obey such laws. No formulae are given here, but, instead, rough charts which enable the necessary pipe size to be fixed. When deciding on the size of a pipe we want to know whether it should be 3 in. or 4 in. (the areas are 7·07 and 12·57 sq. in. respectively). What can be the good of fooling about with $P_1^{1\cdot9375}$?

In deciding pipe size there are three considerations only, apart from cost. What pressure drop along the pipe can be permitted ? What steam velocity can be tolerated ? What is the most suitable stock size ?

If the steam is going to or coming from a reducing valve, there is a certain temptation to use a very small diameter pipe and so allow some or all of the pressure drop to take place in the pipe. There are three objections to this. First, the pressure drop will vary with changes in the rate of flow ; this may give considerable operating trouble. Second, at the very high velocities which accompany a big pressure drop, the steam may quickly erode pipes, especially at bends. Third, steam flowing in a pipe at high velocity, even tenuous vapour at high vacuum, can make an infernal noise.

172. STEAM VELOCITIES IN PIPES. The reasonable speeds for steam of various qualities are fairly closely agreed by most authorities and are shown in Table XXVII.

173. EFFECTIVE PIPE RESISTANCE—EQUIVALENT LENGTH. The charts for finding the correct pipe size are based on pressure drop per 100 ft. of length. Table III, in Chapter I, is constructed for a pressure drop of 1 psi per 100 ft. This is by no means a universally satisfactory basis, so methods of

finding the right pipe size for various conditions will now be given in some detail. To know the drop per 100 ft. the pressure drop over the whole length of pipe that can be permitted must be decided on and this total pressure drop must be divided by the effective pipe length and multiplied by 100. The effective pipe length consists of the actual pipe length plus an addition for each bend, elbow, tee, valve or whatnot. These are given in Table XXVIII.

TABLE XXVII. RECOMMENDED FLUID VELOCITIES IN PIPES

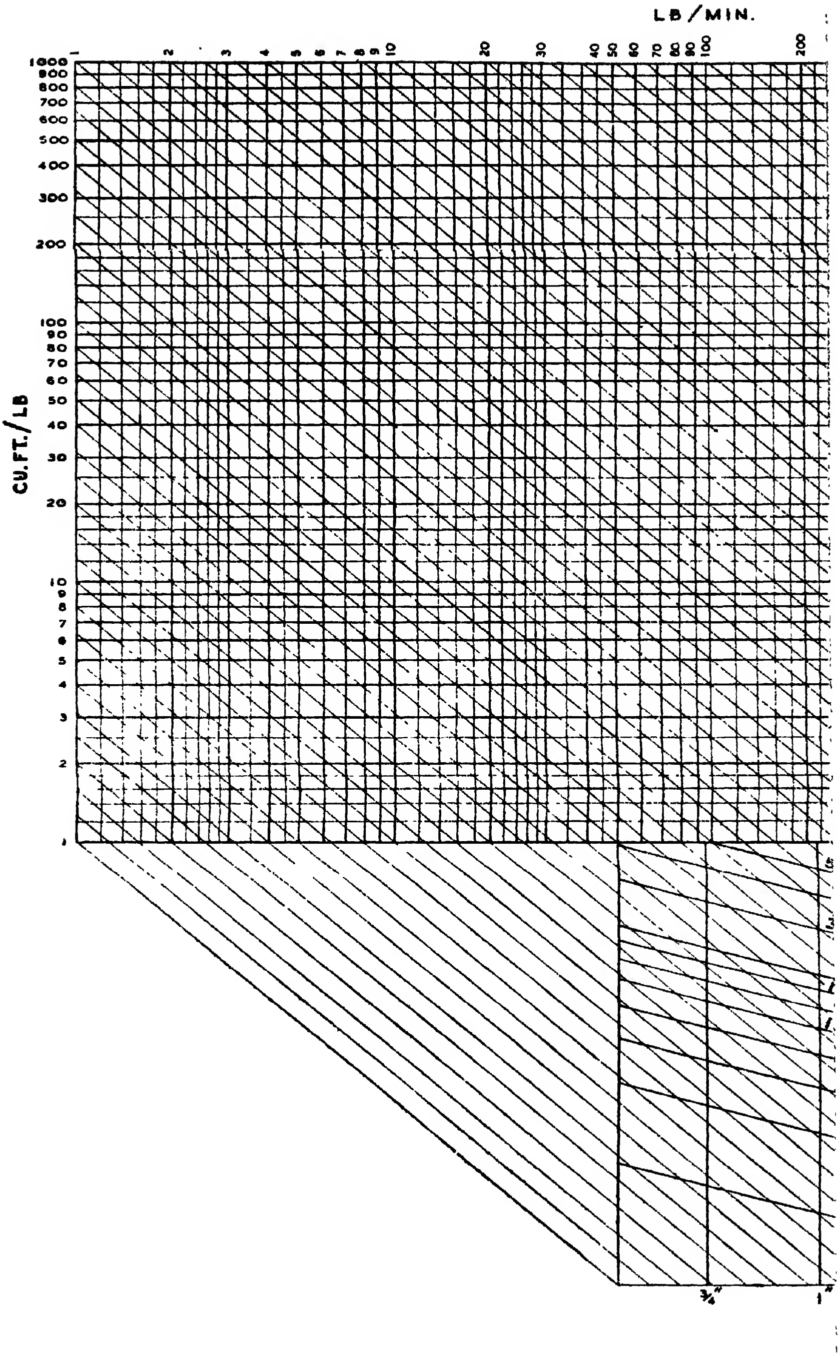
High vacuum water vapour	200–350 feet per second
Moderate vacuum water vapour	150–200 „ „ „
Exhaust steam—wet	70–100 „ „ „
Dry saturated steam	100–130 „ „ „
Superheated steam	150–200 „ „ „
Water	4 – 8 „ „ „

During short heavy peaks higher velocities can be permitted.

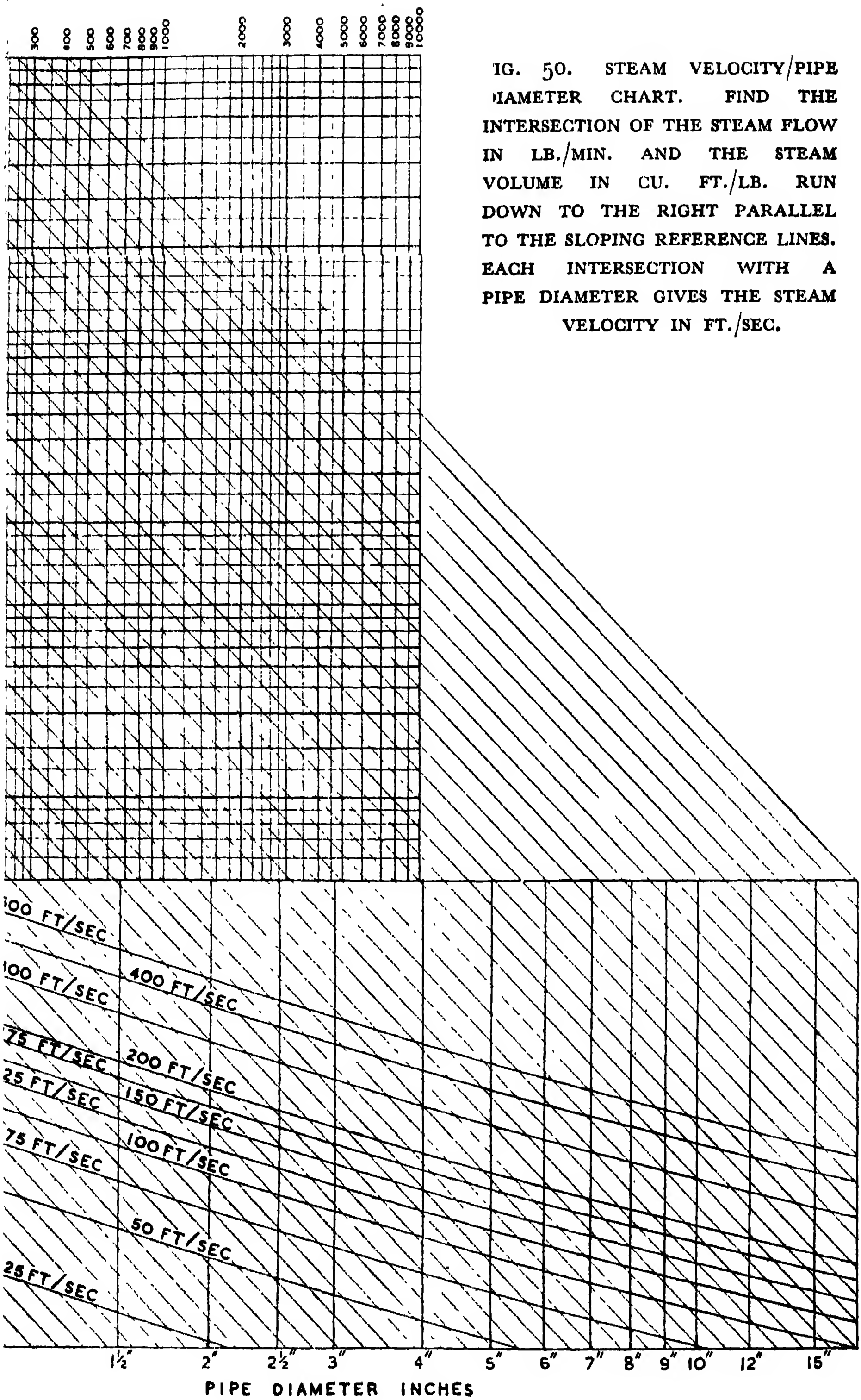
TABLE XXVIII. APPROXIMATE RESISTANCE OF PIPE FITTINGS

PIPE BORE INCHES	EQUIVALENT STRAIGHT LENGTH IN FEET							
	STANDARD ELBOW	STANDARD BEND	TEE		OPEN VALVE			EXPANSION LOOP
			BARREL	BRANCH	SLIDE	ANGLE	GLOBE	
1	2	·5	·5	2·25	·5	1·5	3·25	2·25
1½	3	1	1	3·5	·75	2·5	5·5	3·5
2	4	1·25	1·25	5	1	3·5	7·5	5
2½	5	1·5	1·5	6·5	1·25	4·5	10	6·5
3	6	2	2	8·5	1·5	5·5	12	8·25
4	8	3	3	12	2·25	8	18	12
5	10	4	4	15	3	10	23	15
6	13	5	5	19	3·5	13	29	19
7	15	6	6	23	4·25	15	34	23
8	17	7	7	27	5	18	40	27
9	19	8	8	31	6	21	46	31
10	21	9	9	35	7	24	53	35
12	25	11	11	44	8	30	66	44
15	31	14	14	57	11	34	86	57

Now it is clearly impossible to say what the effective length is until we know the diameter of the pipes, but that is what we are seeking. We must therefore make a preliminary guess. An example will show.



LB/MIN.
1
2
3
4
5
6
7
8
9
10
20
30
40
50
60
70
80
90
100
200
CU.FT./LB
1000
900
800
700
600
500
400
300
200
100
90
80
70
60
50
40
30
20
10
9
8
7
6
5
4
3
2
1
3/4"
1"



IG. 50. STEAM VELOCITY/PIPE IAMETER CHART. FIND THE INTERSECTION OF THE STEAM FLOW IN LB./MIN. AND THE STEAM VOLUME IN CU. FT./LB. RUN DOWN TO THE RIGHT PARALLEL TO THE SLOPING REFERENCE LINES. EACH INTERSECTION WITH A PIPE DIAMETER GIVES THE STEAM VELOCITY IN FT./SEC.

174. USE OF PIPE FLOW CHARTS. Given the quantity of steam in lb./hr., the volume per pound and the effective length of the pipe, the chart, Fig. 49, will show the diameter of pipe needed for any particular pressure drop. Suppose we wish to pass 6,000 lb. of steam per hour at 60 psi.g. along a pipe 450 ft. long round nine standard bends, through the barrels of four tees, through the branch of one tee, round one expansion loop and through one globe valve, with a total pressure drop of 10 psi. We will have to make a guess at the pipe diameter before we can estimate its effective length. Let us guess it as being between 4 in. and 6 in. diameter. The effective length will be :—

	4 in.	6 in.
Straight measured length ..	450	450
Nine bends	27	45
Four tee barrels	12	20
One tee branch	12	19
One expansion loop	12	19
One globe valve	18	29
Effective length	531	582
Pressure drop/100 ft.	1·9	1·7 for a total drop of 10 psi.

The steam table gives the volume of 60 psi.g. steam as 5·84 cu. ft./lb. and at 50 psi.g. (after the drop along the pipe) 6·68 cu. ft./lb. The mean volume is 6·26.

From the charts, Figs. 49 and 50, the correct pipe size can be readily found. These charts may look a bit formidable, but they are really quite simple to use.

In Fig. 49 find the intersection of 6·26 cu. ft./lb. on the vertical scale, with 100 lb./min. (6,000 lb./hr.) on the horizontal scale.

The evenly spaced lines at 63½° are reference lines—(they are actually lines of constant cu. ft./min.^{2}, but that does not matter).

Having found the intersection of volume and quality, move down the reference line to its intersection with a pipe diameter.

The point of intersection of the reference line with the pipe diameter gives the pressure drop per 100 ft. on the vertical scale—in this case

7 psi with a 3-in. pipe,

1·5 psi with a 4-in. pipe,

·45 psi with a 5-in. pipe (our guess of 6 in. is literally off the map).

Clearly the 4-in. pipe is the suitable size. We can now make a closer estimate.

The effective pipe length is 531 ft.

With a pressure drop of 1·5 psi/100 ft. the total drop will be 8 psi,

so that the final pressure will be 52 psi.g.

The mean volume will be 6·17 cu. ft./lb.

This small correction makes no appreciable difference to the reference line that must be used, so that we can take it that the pressure drop along a 4-in. pipe will be 1·5 psi per effective 100 ft. with steam at 60 psi.g. at a flow of 6,000 lb./hr.

The steam velocity must now be found.

In Fig. 50 find the intersection of the volume, 6·17 cu. ft./lb., with the quantity, 100 lb./min.

Locate this intersection on a 45° reference line (these reference lines are lines of constant total volume).

Move down along the reference line to its intersection with the 4-in. pipe.

Read off the steam velocity, 120 ft./sec.

Here is a different approach. Suppose we have an 8-in. pipe that is to carry vapour at 24-in. vacuum ;

that it is 43 ft. long with two bends and one angle valve.

The effective length is 75 ft.

We wish to find out how much vapour can be passed without losing more than ½-in. vacuum.

½-in. Hg. is equivalent to ·245 psi, so that the pressure drop per 100 ft. will be ·326 psi.

Find the intersection of ·326 psi pressure drop with the 8-in. pipe in Fig. 49.

The mean volume of steam at 24-in. to 23½-in. vacuum is 117 cu. ft./lb.

Run up the reference line to its intersection with 117 cu. ft./lb.

This intersection gives the weight as 70 lb./min. or 4,200 lb./hour.

Reference to Fig. 50 shows, however, that the velocity will be 380 ft./sec.

Clearly in this case the pressure drop is not the limiting factor. If we say 250 ft./sec. is the highest vapour velocity that we can accept in this case we find the weight of flow thus :

From the intersection of the 250 ft./sec. line with the 8-in. line run up the reference line to its intersection with 120 cu. ft./lb.

This intersection gives the weight that will pass as 45 lb./min. or roughly 2,700 lb./hr.

The Figs. 49 and 50 separately do not cover the whole range of possible requirements. To do so would have meant impossibly large diagrams, but it is believed that all the necessary data can be obtained from the two figures. Fig. 49 does not cover the very low pressure drops required for very high vacua, nor does the volume scale enable vacua below 27 in. to be used. The volume scale on Fig. 50 does not cover steam above 450 psi but it covers steam down to 29½-in. vacuum. As high pressure steam piping is chiefly governed by pressure drop and as high vacuum steam piping is controlled principally by velocity the whole working range is probably adequately covered. In any case exceptionally high pressures or exceptionally high vacua call for specialists who can well work out the required pipe sizes.

Table XXIX gives a general indication of the relation between pressure drop and velocity for various conditions.

Suppose a steam main splits into branches of different lengths and carrying different quantities to feed different plants. Each small branch, main branch and main pipe must be considered separately ; the flow through each estimated ; the permissible pressure drop decided upon. A balance must then be struck between pressure drop, velocity, cost of pipe, etc. Extra resistance in one section, where a considerable pressure drop may be tolerated, may perhaps be offset by a reduced pressure drop in another section because a convenient length of oversize pipe was in stock.

TABLE XXIX. STEAM VELOCITIES IN PIPES

PIPE SIZE	PRESSURE	VELOCITY—FEET PER SECOND WITH PRESSURE DROP/100 FT. OF :—						
		·25 PSI	·5 PSI	·75 PSI	1·0 PSI	1·5 PSI	2·0 PSI	3·0 PSI
3″ pipe	20″ vac.	135	180	235	270	330	395	500
,, ,,	Atmos.	85	120	145	170	205	250	320
,, ,,	25 psi.g.	50	75	85	105	130	150	190
,, ,,	100 ,,	30	45	55	65	75	90	115
,, ,,	400 ,,			30	35	40	50	60
6″ pipe	20″ vac.	220	315	400	460	550	625	750
,, ,,	Atmos.	145	190	240	300	350	405	500
,, ,,	25 psi.g.	80	115	145	170	200	240	300
,, ,,	100 ,,	50	70	90	105	130	150	180
,, ,,	400 ,,		30	45	55	65	75	95

175. HIGH OR LOW PRESSURE DISTRIBUTION. A steam pipe should carry steam by the shortest route in the smallest pipe with the least heat loss and the smallest pressure drop that circumstances will allow.

For process work we usually want steam at the lowest possible pressure. Apart from greater cost, a big pipe means a large hot surface from which heat can be lost. This is an argument frequently heard. Let us try to find out just how powerful it is. Take a plant raising steam at 60 psi.g. and using it at 15 psi.g. The steam can either be distributed at 60 psi and reduced at the end of the pipe, or it can be reduced as soon as possible, getting the benefit of any wiredrawing superheat, and distributed at 15 psi. Assume that we shall permit a pressure drop at 60 psi of 1 psi/100 ft. and ·5 psi/100 ft. at 15 psi. Fig. 49 shows that, if we wish to pass 15,000 lb./hour we shall need a 6-in. pipe at 60 psi and an 8-in. pipe at 15 psi. Fig. 50 shows that the velocity at 60 psi will be all right but that it will be a little on the high side at 15 psi. Now the steam at 60 psi will have a temperature of 307° F. if it is saturated, and steam at 15 psi wiredrawn will have a temperature of 286° F. (Table VI, Section 48). From Table XXI we see that each pipe will be lagged with 1½ in. of lagging. Fig. 47 says that the 6-in. pipe will lose 76 Btu/sq. ft./hour, and that the 8-in pipe will lose 67 Btu with wiredrawn steam and 56 Btu with saturated

steam. The pipe areas are found in Table XXIV. A tabulation will make things clearer :—

Pressure	60 psi.g.	15 psi.g. wiredrawn	15 psi.g. saturated
Temperature in pipe	307° F.	286° F.	250° F.
Heat loss/sq. ft.	76 Btu	67 Btu	56 Btu
Pipe diameter	6 in.	8 in.	8 in.
Relative areas	1·71	2·23	2·23
Relative heat losses	100	115	96

This shows that by reducing to 15 psi and wiredrawing the heat loss is increased by 15 per cent. This is however not quite the true story because the 15 psi superheated steam will lose heat less readily than it would were it saturated. It has been assumed that the 60 psi steam, before reduction, was saturated and perfectly dry. This is extremely unlikely. If the steam before reduction was more than 2 per cent. wet there could be no superheating by wiredrawing, and the 15 psi steam will be at saturation temperature. This state of affairs is shown in the third column of the tabulation above. We see that there is a 4 per cent. reduction in heat loss if the low pressure steam is at saturation temperature. So that we are probably justified in saying that there is no extra loss, other than higher capital cost, in distributing at low pressure if other requirements make this desirable. (There is a rather unjustified assumption here. Were the steam saturated at 15 psi, the velocity in the pipe would be definitely on the high side. With a 9-in. pipe the heat loss would be just a little higher than the loss from the 60 psi 6 in. pipe.) The approximate cost of piping of various sizes is given in Table XXXI, Section 204.

176. DISADVANTAGES OF SUPERHEAT FOR PROCESS. Superheated steam is generally believed to be bad steam to use for heating processes. For these purposes steam should be dry saturated.

Superheated steam is a dry gas, and it parts with its heat by conduction. When the layer of superheated steam which is in contact with the heating surface has parted with some heat the heat from the body of the steam has to pass by conduction through the very bad conducting gas to reach the outer layer. Saturated steam gives up its latent heat by contact with the heating surface. When the contact layer of steam has condensed more steam flows towards the surface to take its place. The heat does not have to flow. The steam flows. There are few published figures of the difference between the heat transfer rates from saturated steam and superheated steam. Some people suggest that the higher temperature of superheated steam compensates for its lower rate of heat transfer. Others maintain that in spite of the higher temperature the rate of heat transfer is much lower.

A brewery in Eire used in its coppers steam that came direct from the superheater of the boiler. The insurance company condemned the superheater so that wet steam was fed to the copper with much foreboding. The head brewer was surprised and delighted at the great improvement in the rate of boiling under the new conditions with wet steam.

A board drying machine in the South of England consisted of a wide tunnel in front of which was a bank of horizontal steam heated pipes. The drying air was blown through the heater and down the tunnel. Steam entered the left-hand manifold of the heater and condensate was drained from the right-hand manifold. The steam used had 100° of superheat. The performance of the drier was quite unsatisfactory. When the boards in the right-hand side of the tunnel were correctly dried, those in the left-hand side were quite damp. This proved that although the steam in the left-hand side of the air heater was 100° hotter than the right-hand side, the superheated steam parted with its heat so reluctantly that it did not heat the air so readily as did the lower temperature saturated steam on the right-hand side.

On the other hand a laundry near London reports better drying performance with superheated steam than with saturated steam.

It can be shown in industrial practice that superheated steam used inside a heating surface of calandria type gives just as good heat transfer as saturated steam. It is suggested that in heating surfaces of this type the steam comes into almost immediate contact with condensate and desuperheats itself by evaporating some of the condensate.

The theory that heat transfer is slower from superheated steam than from saturated steam rests on rather rickety foundations. If, in fact, the metal of the heating surface in contact with superheated steam is not coated with a film of condensate the metal must be above saturation temperature and the heat transfer must be greater. There is possibly some confusion between heat transfer coefficient per ° temperature drop and total heat transfer.

Whatever the truth about heat transfer rate, there are several other objections to using superheated steam for process. The first is that the temperature of the heating surface is not definite. The bulk of the heat transfer takes place at saturated temperature but some of it may take place at some higher temperature. This may be undesirable if a material is to be heated up and then maintained at a particular temperature. During the heating-up process the heat transfer is rapid and most of the heat transfer will take place at saturation temperature. When the material is up to temperature and the steam supply is reduced a large part of the heating surface may be at superheat temperature, and, if the circulation of the process material is sluggish, there may be local overheating with damage to the product. Saturated steam has a definite upper limit of temperature. With superheat the upper range of temperature of a heating surface is not really known. It probably will not be equal to the steam temperature, but it may well be considerably above saturation temperature.

Where the material to be heated is a delicate product, easily damaged by overheating, superheated steam must be avoided.

Another disadvantage of superheated steam for process is that large temperature stresses may occur in the plant. If superheated steam is fed to a calandria vessel the calandria is near saturation temperature, and the steam pipe is at superheat temperature. The flange does not know whether it is coming or going.

If superheated steam is used for direct injection into a liquid, before it can condense it must give up its superheat. If the liquid depth is shallow, three or four feet—a common depth in many vats or becks—there is very little chance

that the superheat will be given up and condensation completed in the short time that the bubbles take to pass up through the liquid. Much of the steam may reach the surface uncondensed and escape.

There are rare cases where superheated steam is passed rapidly through a piece of plant and gives up superheat only to some high temperature process; the steam then passes on to some more orthodox process plant where condensation takes place. By rapid passage through the special plant heat transfer is more rapid, the temperature gradients are smaller and the high temperature maintained.

It can be said therefore that superheated steam may have definite disadvantages for ordinary process uses. For direct injection it is unsuitable. If, however, it is convenient to use it the result may be perfectly satisfactory.

There is one advantage which superheated steam has owing to the fact that each pound of steam contains more heat units. For a given heat input there will be less condensate. Where it is not possible to collect flash steam from the condensate, this reduces the heat loss.

177. DISADVANTAGES OF WET STEAM. Wet steam is ready to condense and give up its latent heat immediately it touches a cooler surface, but the moisture that it carries is just a nuisance. It puts unnecessary condensate on to the heating surface. The moisture contains no available heat. The trapping and condensate system has to deal with a lot of unnecessary water. The extra condensate produces extra flash steam which must be collected, or, if this is not possible, the extra flash is just extra loss.

178. ADVANTAGES OF DRY SATURATED STEAM. Dry saturated steam is, like wet steam, all ready to condense at once on a cooler surface and give up its latent heat. It yields no unnecessary condensate. Its temperature is definite and constant. It is in the ideal state for use in process or in heating plant.

179. STEAM QUALITY AS PRODUCED. Steam from a boiler without a superheater is seldom or never dry. Steam from a Lancashire boiler has been found on occasion to be as much as 20 per cent. wet. Exhaust steam from an engine or turbine can be either wet or superheated, as has been explained in Chapters 2 and 3. Steam should always leave the boiler or power house with sufficient superheat to reach the process dry.

180. STEAM QUALITY AND DISTRIBUTION. The ideal steam distribution system should bring steam to the plant at dry saturation point. This means that steam must leave the boiler house or back-pressure machine with just sufficient superheat to enable it to reach the plant without wetness and without superheat. This is an ideal that cannot be reached in practice. A compromise must be made. The largest process heat user should be taken and the steam condition so arranged that this largest plant gets dry saturated steam. Plant on the supply side will get superheated steam.

Let us take the case of the pipe whose diameter was found in Section 174.

This 4-in. pipe is carrying 6,000 lb./hr. steam at 60 psi.g.

The saturation temperature is 307° F.

The radiation surface is, with flanges, etc., about 620 sq. ft. (Table XXIV, and Section 190).

The heat loss is about 78 Btu/sq. ft./hr. (Table XXIII and Fig. 45).

The total heat loss is $620 \times 78 = 48{,}360$ Btu/hr., or $8 \cdot 1$ Btu/lb. of steam.

A superheat of 20° F. should give a small margin and be quite sufficient.

181. STEAM DRIERS. There are many steam driers—so-called—on the market. Many of them are not much use. The separation of minute water droplets from fast flowing steam is not an easy matter. The most serious attention has been given to steam drying in the drums of high pressure boilers and between the intermediate and low-pressure cylinders of large turbines. In some boiler drums drying is done by washing the steam with some of the circulating water. Such methods are hardly applicable to steam pipes. Some of the turbine steam driers are said to reduce the moisture content down to ½ per cent. Nothing approaching this can be expected from the ordinary baffled pot which is such a familiar illustration in catalogues.

Many driers work on the centrifugal principle. The steam is given a swirling motion in the outer large diameter part of the drier. It is then made to turn through 180°. It maintains its rotary motion and as it passes up the smaller central section its rotational speed increases in order to maintain its angular momentum. Fig. 51 shows this type of drier. The moisture is thus thrown on to the outer surfaces, down which it runs to the drain outlet.

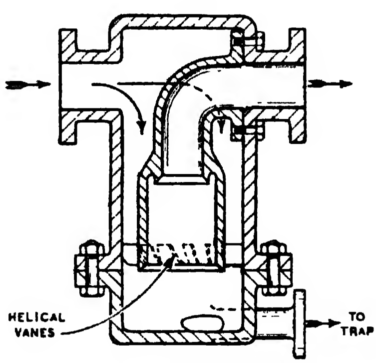


FIG. 51. STEAM DRIER

The ordinary baffled pot depends also on centrifugal force for its action. It is the centrifugal force acting on the heavier water droplets at the change of direction of the steam round the baffles that throws them outwards. The faster steam is flowing the greater will be the centrifugal force, consequently the faster the flow through the drier the greater the drying effect. There is however a limit. If the flow is too fast the deposited water will be brushed off the sides

of the separator and will be carried on with the steam. Steam driers therefore are not at all adaptable to fluctuating loads. If the flow is too slow there will be little or no separation. If the flow is too fast, much of the good separation may be undone. The size of a steam drier therefore must be chosen with care. A margin "for luck" just makes certain that the thing cannot work properly.

There are other types of steam drier which do not rely upon centrifugal action. In some types the steam passes plates which carry projections which collect the water droplets. Driers of this kind are said to do their work over a wider flow range than the centrifugal types.

182. WIREDRAWING EFFECT. Suppose the process requires steam at 30 psi.g. and suppose it is raised in the boilers at 100 psi.g. Suppose also that it is saturated and that it is distributed at 100 psi. It will part with its heat readily all along the line with resultant condensation. Much of this condensation will reach the pipe wall and will be removed by the pipe draining system. When the 100 psi steam reaches the point where it is to be used it is reduced in pressure to 30 psi. It will be wiredrawn and superheated (Section 48).

Saturated steam at 100 psi.g. contains 1,191 Btu,

whereas saturated steam at 30 psi.g. contains only 1,173 Btu.

The pressure reduction makes 18 Btu available for superheating the 30 psi steam by 36° F. (A good rough rule for low and medium pressure steam is to take the specific heat as being ·5, so that 1 Btu will raise the temperature 2° F.)

If the steam contains moisture there will be no superheating. The excess Btu will be used in evaporating moisture.

The latent heat at 30 psi is 930 Btu, so that the 18 Btu will evaporate

$$\frac{18}{930} = \cdot 019 \text{ lb. of water or } 1\cdot 9 \text{ per cent. of wetness.}$$

If the steam contains less than 2 per cent. of moisture this will be all very nice. Unfortunately, the steam is much more likely to be 5 per cent., 10 per cent. or even 20 per cent. wet. So we see that this last minute drying by wiredrawing is rather a myth unless the pressure reduction is very great, in which case some better use should be made of the pressure drop than just saving the face of the entropy increase by calling it drying.

It is sometimes suggested that by lowering the pressure at the earliest possible point and doing the bulk of the distribution at low pressure the steam will be wiredrawn and dried. It is now clear that this can only be a good argument if the steam was pretty dry to start with. But it is true that the steam will probably be drier at the beginning of its journey than near its end and, provided the steam mains are large enough, it is better to do as much early pressure reduction as possible and take advantage of the superheat by wiredrawing however small it may be.

183. SUPERHEATING AND LOW PRESSURE. Steam should leave the boiler slightly superheated and should, if possible, be produced at only a few pounds above the process pressure—just sufficient excess pressure to compensate for the pressure drop along the pipe. Similarly, engine or turbine exhaust

should be arranged, if possible, to be slightly superheated. This is easy on a turbine but reciprocating engines must be lubricated if they are to use superheated steam and this will almost certainly call for some device to remove the oil from the engine exhaust.

Where the whole of the steam does not pass through the power-producing plant and where the live steam is superheated, the steam reduced direct from the high-pressure main for augmenting the engine exhaust may have enough superheat in it after wiredrawing to dry the exhaust from the engine with which it mixes. Where the amount of reduced steam is relatively large compared to the exhaust there may even be too much superheat in the mixed process steam.

It becomes more and more apparent that, apart from first cost, all arguments seem to favour the distribution of steam at the process low pressure. There may be difficulties about getting the right amount of superheat. If the process steam is reduced from high pressure, especially from superheated high pressure, it may be too hot at the point of use. This will call for desuperheating. If reciprocating engines are used with initially superheated steam and exhausting to process, oil separators will be needed.

184. OIL SEPARATORS. Oil cannot be permitted to come into contact with most process products ; for example, food, drink or textiles. If steam is blown into any process vessel direct it must be oil-free. Even if no steam is used for direct injection oil is liable to coat heating surfaces and reduce heat

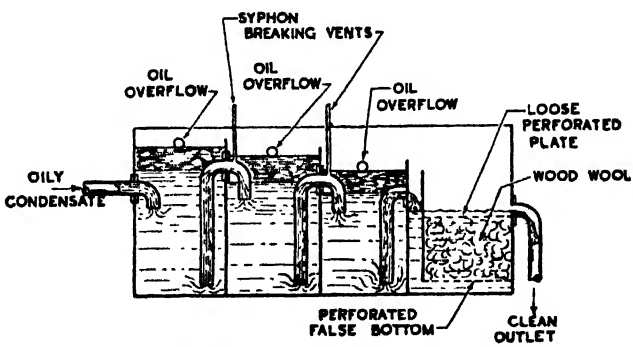


FIG. 52. OIL SEPARATOR FOR CONDENSATE

transfer. Oil is a very bad conductor of heat. Some published figures show it as being ten times as bad as boiler scale. If the condensate is used for boiler feed it must be free from oil.

Oil separators are made for removing oil from steam or from condensate. There do not appear to be any completely satisfactory oil separators. Possibly it might be said that removing oil from steam is a bit easier than trying to remove it from condensate, but this is only true if a very low standard is set.

It is possible to remove almost all the oil from condensate with a somewhat elaborate apparatus, such as is shown in Fig. 52. The condensate is run into a series of vessels, the outlet from each coming from the bottom with a slow flow so as to give the oil plenty of chance to float to the surface, from which it can overflow. This however will never remove the last trace, which must be taken out by filtering the water through sand or wood wool, either of which need frequent replacement.

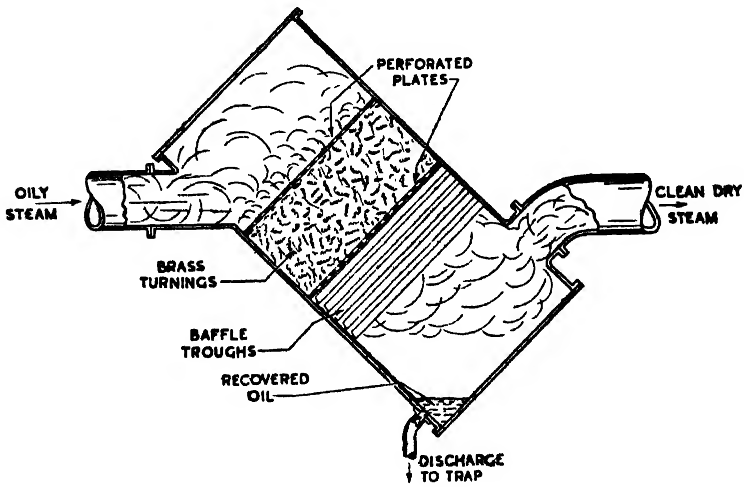


FIG. 53. OIL SEPARATOR FOR STEAM

Fig. 53 shows an oil separator designed to take the oil out of steam. The theory is that by making the steam pass over a large surface the oil will be deposited. Some separators are just vessels full of fine chains on which the oil can stick. None seem to be perfect.

If the engine is exhausting into a process main and the power demand is such that the engine efficiency is not of great importance, it will be better to use saturated steam and avoid the difficulties of lubrication and exhausted oil. If, however, the steam/power situation is such that every effort must be made to get every last little bit of efficiency from the power plant, then superheat should be used and the oil difficulties tackled as well as possible.

If oil is present in steam as vapour, as it may be if the engine in which it was used took very superheated steam, oil separators of the kind shown in Fig. 53 may not be satisfactory.

If oil is present in condensate in the form of an emulsion the oil droplets are so small that they will not coalesce into larger drops and separate of their own accord. Oil separators of the kind shown in Figs. 52 and 53 are seldom 100 per cent. efficient.

In many cases, especially where little oil is present and with saturated steam, fabric filters are used. These are often quite satisfactory, but must be cleaned frequently and the cloth or blanket replaced regularly.

To remove all the oil, condensed oil vapour and emulsified particles, requires other means. The principal methods are chemical. Reagents are added which form a precipitate of aluminium hydroxide or sodium aluminate. These precipitates entangle the oil droplets which can be filtered off. Information about complete oil removal from condensate can be obtained from firms selling water treatment plant.

The removal of oil from boiler feed water is usually looked upon as of first importance. Oil in a boiler can cause foaming and priming, tube or plate overheating, and sometimes corrosion. Oil also tends to bind together sludge particles and encourages troubles in superheaters and deposits on turbine blades.

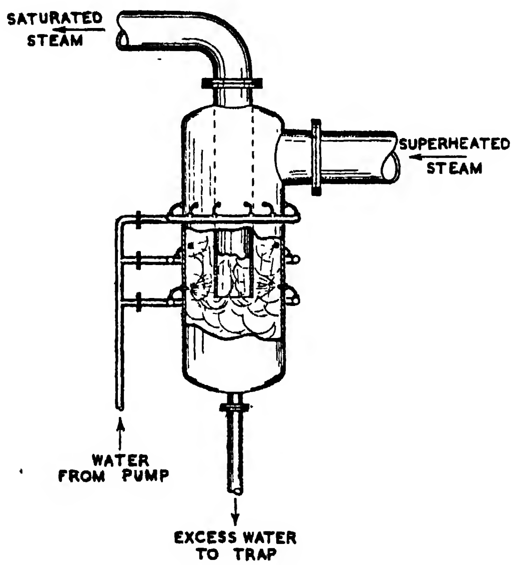


FIG. 54. SATURATED DESUPERHEATER

185. DESUPERHEATERS. Where steam reaches the process plant with much superheat it may have to be desuperheated. This was briefly referred to in Section 49. Desuperheaters can be of three kinds. The first two employ a spray of water through which the steam passes. The superheat evaporates some of the water and the heat that was present as superheat is changed into latent heat. If an excess of water is sprayed into the steam complete desuperheating will take place and the steam will emerge, not only saturated, but to some extent wet. The water spray can, however, be controlled by a thermostat so that the partly desuperheated steam is held at a fixed temperature. In the case of all these contact desuperheaters the water used must be pure distilled water. If water contains any solids they will be thrown out and will quickly scale up the desuperheater.

Fig. 54 shows a simple desuperheater that takes all the superheat out. An excess of water is pumped through the sprays, the excess returning to the pump from the outlet at the bottom through a trap.

Fig. 55 shows a thermostatically operated desuperheater that is inserted into the steam pipe. In this device the size of the vortex spray nozzle is controlled by the thermostat rotating the segment which can close or open the throat

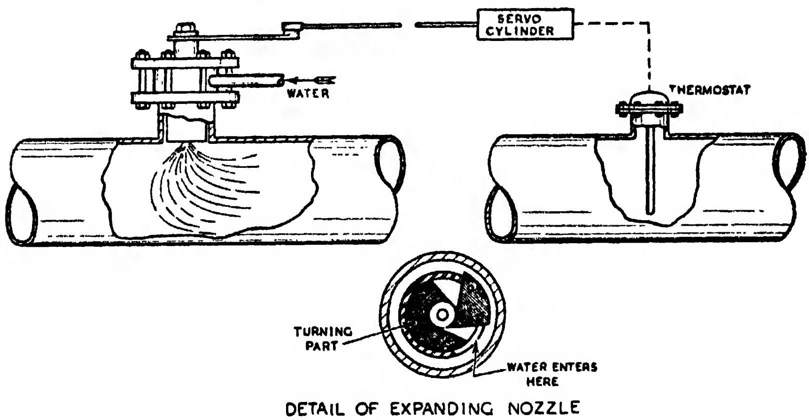


FIG. 55. THERMOSTATIC DESUPERHEATER INSERTED IN PIPE

of the vortex. This ingeniously gets over a major difficulty. Any form of spray nozzle ceases to spray and becomes a dribble when the flow through it is reduced to a certain point. The device shown in Fig. 55 is an expanding spray nozzle and continues to spray down to very low flows.

In Section 146 one of the requirements of the plant under consideration was that the quantity of steam was to be a minimum. Any desuperheating by contact with water must increase the quantity of steam. In the example in Section 146 this increase was 8·95 per cent. In order to desuperheat without increasing the steam quantity it will be necessary to use a dry desuperheater, in other words a plain heat exchanger or calorifier. The chief difficulty of using such a device is to find a suitable liquid to receive the heat addition. The temperature may be very high, and it is waste of good heat potential to heat cold or warm water. It might be possible to take the water leaving the economiser and heat it on its way to the boiler, but this will leave some extra heat in the flue gas and will call for air heaters lest what has been gained on the desuperheating roundabout is not lost on the flue-gas swings. In some factories where oils or other high boiling point liquids are dealt with there may be scope for dry desuperheating, but in most factory processes the heating of process liquors can be largely done with low-grade heat. Heating by desuperheater is merely a disguised way of heating by live steam.

186. MEASURING STEAM QUALITY. All this talk of dry steam, wet steam and superheated steam is rather beside the mark if we do not know whether steam at any particular part of the plant is dry, wet or superheated.

If steam is superheated a thermometer in the pipe should show this correctly. Suppose the pressure in the pipe is measured on a gauge whose accuracy has been checked and is found to be 24 psi.g. The steam table gives the saturation temperature of steam at this pressure as 265° F. If the thermometer, which must also be checked for accuracy, shows 295° F., there is clearly a superheat of 30° F. equivalent to a superheat heat content of 15 Btu.

If the steam is wet the measurement is much more difficult. It can be said at once that a really accurate measurement is impossible, but with care a result quite near enough for practical factory purposes can be obtained. The difficulty is not in the measurement ; it is in the almost impossibility of being certain that a representative sample of the steam flowing along the pipe has been taken. A description of the way in which wetness in steam is measured is given in the next chapter.

The quality of steam should be measured at a number of different points in the factory. The amount of moisture in most factory steam is not generally appreciated. Could one but pay a visit to the inside of our steam pipes one would only too often be enveloped in a heavy Scotch mist. A large factory in Scotland (appropriately) recently found the steam at the end of one of its distribution lines to be 75 per cent. wet.

187. DISTRIBUTION LAY-OUT. The arrangement of steam pipes—any pipes in fact—is in most factories lamentable. Factories are generally laid out with a view only to plant convenience or to making the best use of an unsuitable building. Often the piping is not designed at all. It is " coupled up " after the plant is in position. As the years pass alterations and additions are made so that most factories a generation or so old have a tangle of bends, tees, elbows, valves, cross-connections, bye-passes, dead-ends, etc., such that each piece of plant passeth all understanding.

Piping calls for just as much care and thought as any other part of the plant design. Actually it is often much more difficult to design good piping than to design good plant. Every unnecessary bend, tee, elbow or whatnot should be eliminated—all they do is to increase entropy. Kinks and dogs' hind legs mean either a larger pressure drop or a larger pipe or a restricted flow. When vertical and horizontal pipes are carried on the same wall, only too often does one pipe jump another by means of four elbows. In a 4-in. pipe this is equivalent to 32 ft. of extra pipe length on the resistance to flow—see Table XXVIII. A good rule to make is that all vertical pipes be fitted snug against the wall, and that horizontal pipes be carried a foot or so clear of the wall. This avoids the need for jumping and gives the greatest floor space, as horizontal pipes can generally be carried above head level.

While pipes should always be as short as possible, short pipes mean rigid pipes and troubles due to expansion may occur, with cracked or damaged flanges—see Sections 192–202.

Elbows should never be used if there is room for bends—the easier the bend the better. Tees should be replaced with easy Ys. Unfortunately, easy Ys are not stocked by most pipe fittings makers except in small low-pressure sizes. Unless there is some good reason for using them, globe valves should not be

used in preference to slide- sluice- or gate valves. With a little care and trouble it is often easy to halve the resistance offered by a pipe. This may enable a smaller pipe to be used or may enable the pressure to be reduced.

Distribution lay-out may be inextricably mixed up with other things. We shall see later that steam pressure has a marked effect on heat output in a piece of plant, as also has the proper draining of condensate.

In the carding room of a certain West Country woollen mill the combs are heated by steam which enters the first comb heater at 90 psi. The outlet from this heater goes straight to the inlet of its neighbour and so on in series for eight combs. The first comb gets good fairly dry steam at 90 psi ; the last comb gets seven parts of condensate and one part of steam at about 60 psi. The management insisted that exact temperature was of supreme importance, and that long experience had proved that 90 psi was essential. Actually all the combs are at different uncontrollable temperatures. The correct lay out for this carding plant is, of course, a main steam pipe with a branch to each comb whose heater should be individually trapped. Then 20 psi would probably prove as effective in Somerset as it can be in Yorkshire.

Where pieces of plant are large steam users, or where pipe branches feed large groups of small users, an endeavour should always be made to arrange part of the pipe feeding them so that there is a straight length for at least 20 pipe diameters, preferably 30 diameters. This will enable an orifice for a steam flow meter to be fitted with the assurance that its indications will be accurate.

188. IMPROVING OLD PIPING TANGLES. Much can be done to old lay-outs with much benefit. Pipes often go round three sides of a building instead of along the fourth side. Methods change, plant changes, loads change, but the old pipes are modified, tinkered and made to do. By re-arranging old piping there are generally three benefits : a great saving in heat loss ; a material reduction in pipe resistance ; a lot of spare piping to put in store.

189. DEAD PIPES. When a steam pipe is put out of use, this is generally done by one of two methods, both of them bad. The valve controlling the supply to the branch that is to be shut down is closed, or a slip joint (or blanking disc) is slipped into one of the pipe joints. If the plant is going to be dead for only a few days these methods are all right, but if it is to be laid off for months or indefinitely something better must be done. Valves can and often do leak. A slip joint after a few months can rust through without showing any outward sign. So-called cold piping is often piping hot.

Steam pipes that are to be made dead for weeks or indefinitely should be properly put out of use. There is only one proper way. Remove a length of pipe, an elbow or a bend as far upstream as possible and fit a blind flange, or a plug to the live branch. If a slip joint must be used, it should be made of stainless steel or monel metal.

190. DRAINAGE OF PIPES. Even if superheated steam is being used there will always be condensation when the pipe is cold at every start-up and shut-down, whether these occur daily, weekly or yearly. Provision for draining away water must always be made. The water may not be due to condensation ; it

may be carry-over from the boiler. All pipes should be erected with a slight fall in the direction of flow. Draining points often receive even less attention than pipe lay-out. Drains are often added haphazard after the pipe has suffered from severe water hammer. Drainage should be considered with the whole piping arrangement and should receive as much attention as any other requirement.

Fig. 56a shows a common method of fitting a pipe draining connection. Fig 56b shows an even worse but not uncommon way. In Fig. 56b the pipe cannot possibly drain. In Fig. 56a very little condensate is likely to find its way out. Table XXVII gives the recommended speed for saturated steam as

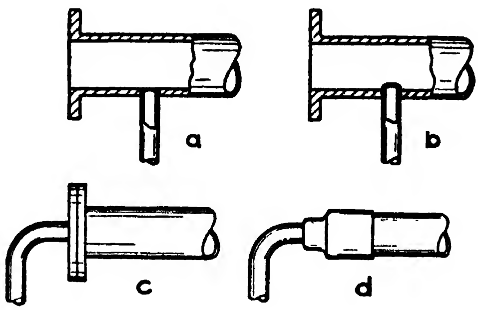


FIG. 56. WRONG WAYS OF DRAINING PIPES

100 to 130 ft./sec. or 70 to 90 m.p.h. The condensate is being dragged along at perhaps 30 m.p.h. and nothing like a $\frac{3}{4}$in. hole in 4 in. pipe will catch it. The draining of a blind end in any such way as shown in Fig. 56c and 56d just makes sure that the pipe will be quarter full. To drain a steam pipe properly there must be proper catch pots at regular intervals. Fig. 57a shows the proper way of draining a straight horizontal pipe. Fig. 57b shows a method of draining a vertical bend. This is good for steam flow ; not quite so good for condensate catching. Fig. 57c shows the right way of draining a vertical elbow, by using a full bore Tee. This is hard on steam flow, but excellent for catching condensate. Figs. 57d and 57e show the right way of draining a blind end. The arrangement shown in Fig. 57f has much to recommend it. By taking the trap connection from a point an inch or two above the bottom of the Tee a good dirt trap is provided for the collection of grit, scale, etc.

There are other very important considerations about the draining of steam mains. They ought perhaps to be dealt with in the Chapters on condensate handling and traps, but they are so important that they will be discussed now and reference can be made to Chapters 8 and 9 if necessary.

The draining of a steam main is particularly important when it is being warmed up from cold. The conditions are not always properly appreciated, with the result that much damage from water hammer (see the next Section) can be done. The higher the pressure for which the piping is designed the greater are the difficulties. Often piping is installed where clearly there has been no attempt to estimate the conditions that will occur during warming up, and the trapping system is little more than a gesture.

High pressure traps (see Chapter 8) have a very small outlet valve. This is for two reasons. First, in a high pressure trap there is a very high pressure difference across the valve opening and consequently the flow will be very fast. Second, the mechanism of the trap has to open the valve against the working pressure ; if the valve had a big area this would mean that the operating mechanism would be very large, powerful and clumsy.

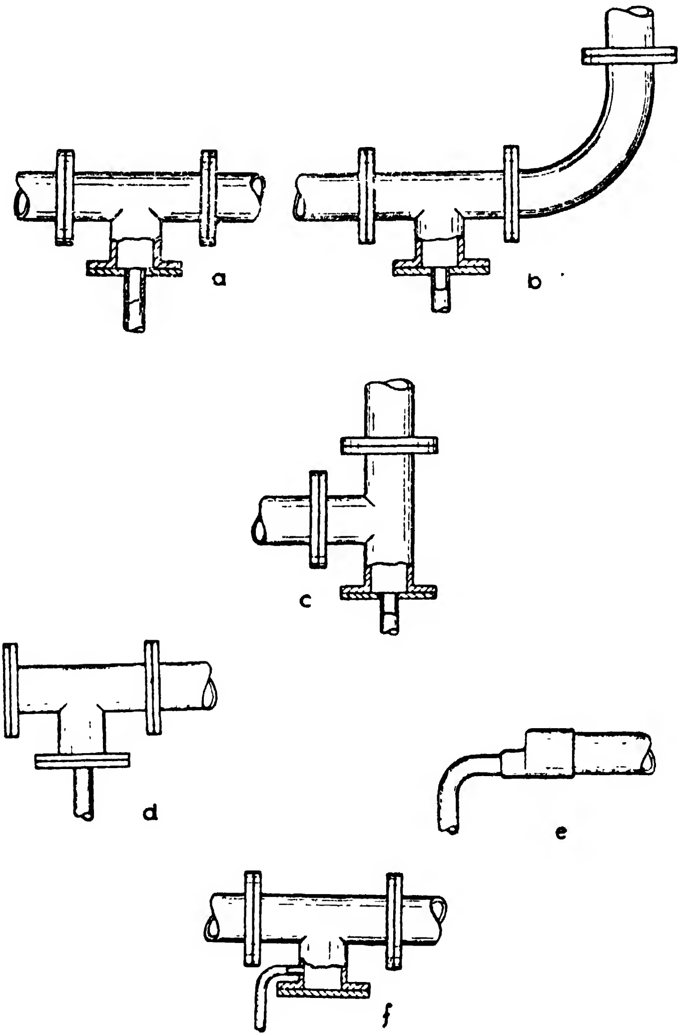


FIG. 57. RIGHT WAYS OF DRAINING PIPES

If the pressure drop across the small valve of a high pressure trap is very low, very little condensate will be discharged. When the pipe is cold and steam is first turned on condensation will be very rapid while the heat transmission to the cold pipe is fast. At the same time the pressure in the pipe will be low,

partly because the cold pipe will be acting as a condenser and partly because ordinary prudence discourages most people from quick raising of pressure in a high pressure main. Consequently during the time when there is the most condensate to get rid of, there is the least pressure to discharge it. Matters are sometimes even worse. In many factories the condensate is returned to a level above the steam pipes. This extra head, together with the low pressure during warming up may prevent the discharge of any condensate whatsoever.

In many plants it is common practice to blow direct to atmosphere during warming up, by opening the trap bye-passes. This is a wasteful practice because the bye-pass is often left open much too long for safety's sake. If the draining system is not quite perfect, the bye-passes should be opened and the drainings blown to atmosphere. It is far better to waste a little steam and condensate than to induce water hammer. There should be no need to open bye-passes ; no need for water hammer ; but the draining arrangements must be properly done.

(*a*) The traps must be as far below the steam pipe as possible. This gives the condensate a chance to add its own weight to help it out when the pressure is low. If, for example, the traps could be 30 feet below the pipe the condensate would exert a hydrostatic head of 13 psi on the trap discharge valve.

(*b*) Condensate, drained from a steam *pipe*, should not be discharged to a higher level. The trap is then relieved of all back pressure and the low pressure inside the pipe together with the hydrostatic head are enabled to do their unimpeded best to drain the pipe.

(*c*) The condensate collecting pockets should be generous. This provides safety reservoirs.

(*d*) The traps should be large enough to discharge the warming-up condensate at warming-up pressure at a sufficient rate to prevent the condensate collecting pockets from filling right up.

Trap discharge rates under various pressures can be obtained from the makers. The amount of condensate made during warming-up can be very easily estimated.

The specific heat of iron is $\cdot 15$, so that 1Btu will raise 1 lb. of pipe $6\cdot 66°$ F. The heat available for warming up is the latent heat at the actual pressure inside the pipe.

So that the warming up of 1 lb. of pipe will condense

$$\frac{\text{Temp. Rise} \times \cdot 15}{\text{Latent Heat}} \text{ pounds of steam.}$$

Take, as an example, the 4 in. pipe considered in Section 174.

Its straight length is 450 feet.

It has 16 fittings, each with a pair of flanges.

There will perhaps be another 40 flanges in the straight pipe.

Table XVIII in the Appendix gives the weight of this pipe as $9\cdot 7$ lb./ft. and $12\cdot 7$ lb./pair of flanges.

This makes a total weight of some 5,100 pounds.

The temperature of 60 psi.g. steam is 307° F.,

and the latent heat is 905 Btu.

If the pipe is to be heated from 60° F. the temperature rise will be 247° F.

The amount of condensate produced to warm up this pipe will be

$$\frac{247 \times 5{,}100 \times \cdot 15}{905} = 209 \text{ lb. or } 21 \text{ gallons.}$$

At first the pressure will not rise to 60 psi so that the latent heat will be higher. While the pipe is cool much of the sensible heat in the condensate will go to heat the pipe. Both these will reduce the actual weight of condensate produced. On the other hand the pipe is losing heat all the time by radiation and convection, which increases the amount of steam needed. Roughly and readily the foregoing estimate is quite near enough.

This pipe will have a surface of about

$$(475 \times 1 \cdot 178) + (56 \times 1 \cdot 134) = 623 \text{ sq. ft. (Table XXIV).}$$

If it is lagged with $1\frac{1}{2}$ in. lagging it will lose (Table XXIII) 75 Btu/sq. ft./hr.

The amount of condensate will be $\dfrac{623 \times 75}{905} = 52$ lb. or $5 \cdot 2$ gall/hour.

If the pipe takes 5 minutes to warm up the draining system will have 50 times as much condensate to handle during warming as during running ; and, during warming, there is much less inducement for the condensate to leave.

191. WATER HAMMER. Water is to all intents and purposes incompressible. Normal steam flow is at about 80 miles an hour. If water, condensate or boiler carry-over, collects in the bottom of a pipe, such as the sorry sagging system shown in Fig. 58 ripples will form on the surface. The steam will blow these ripples into waves until a wave is high enough to fill the pipe. There is then an incompressible liquid piston travelling along the pipe at steam speed.

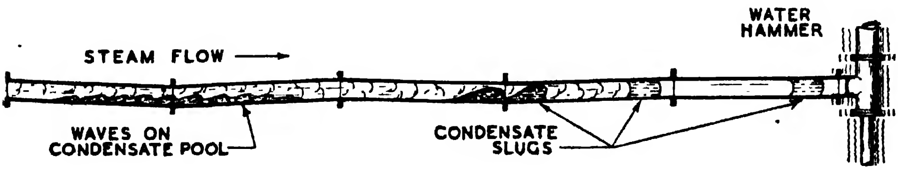


FIG. 58. THE INITIATION OF WATER HAMMER IN A BADLY DRAINED PIPE

When this slug of water, travelling at 80 m.p.h. reaches an elbow, valve or whatnot, it is brought up—bang. Water hammer may be dangerous. Slight hammer can be harmful. It can damage traps, make joints leak, damage thermostat bellows, etc. Bad hammer can fracture pipes. Water hammer must never be tolerated. If it is due to the condensing of steam during distribution in

steam pipes or if it is due to boiler carry-over it can be completely eliminated by giving the piping a proper fall and arranging proper catch pots and providing proper sized traps and drains.

Water hammer in steam pipes occurs for quite a different reason from the hammer that sometimes happens in an economiser or hot water pipe. If the water in an economiser is brought up to the boiling point appropriate to the particular pressure, any sudden demand for water from the boiler will cause a momentary drop in pressure and some of the water will flash into steam. If some cooler water—cooler by a very few degrees—reaches one of these pockets of steam, and particularly if this is accompanied by a small rise in pressure, the steam will condense and the water will come together with a bang.

192. EXPANSION. The amount of expansion that occurs when a steam pipe is heated up from cold is well known, but it is often forgotten. Table XXX sets it out as a reminder.

TABLE XXX. THERMAL EXPANSION OF PIPES

TEMPERATURE RISE °F.	EXPANSION IN INCHES PER 100 FEET
From 60 to 150	·75
60 200	1·15
60 250	1·6
60 300	2·0
60 350	2·4
60 400	2·9
60 450	3·3
60 500	3·8
60 600	4·8
60 700	5·8
60 800	6·9
60 900	8·0

Some kind of provision is often made to accommodate expansion, but seldom are proper steps taken to see that the expansion movement goes the way it was intended to by providing proper anchorages for the pipes. Before considering anchorages there are certain points about expansion devices that must be gone into.

There are a number of different arrangements in common use for taking care of the expansion movement in pipes. Some of them differ very considerably from the others in qualities and behaviour. The following list gives the more important devices :—

(1) The sliding expansion joint.
(2) The expansion bellows.
(3) The expansion diaphragm.
(4) The lyre loop.
(5) The full loop.
(6) The creased bend.
(7) Plenty of length and bends.

193. THE SLIDING EXPANSION JOINT. The sliding expansion joint is shown in Fig. 59. The construction is quite clear. It consists of a socket carrying a packed gland in which the other part of the joint can slide. When pressure is applied to the inside of the joint it forces the two portions apart; so provision must be made by means of the retaining bolts shown to prevent the whole thing blowing open. The steam pressure acts, not on the cross sectional area of the pipe but on the cross section of the sleeve part of the joint—a considerably bigger area. At 260 psi the force pushing in either direction trying to blow the joint apart is about $1\frac{3}{4}$ tons on a 4-in. pipe. Unless the pipe anchorages are properly looked after, these joints generally blow out to the full extent that the retaining bolts will permit (the position shown in Fig. 59). As the pipe warms up and expansion takes place the expansion is not taken up

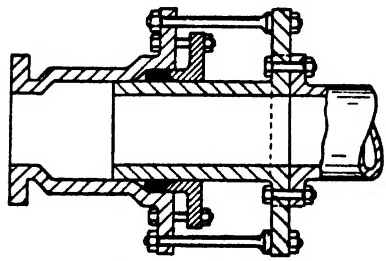


FIG. 59. SLIDING EXPANSION JOINT

by the expansion joint; the pipe just moves further outward. Unless the anchorage is really looked after, the net disturbance to the pipe system can be greater with a sliding expansion joint than with no provision for expansion at all. If the pipes are properly anchored in the proper place to prevent the joint blowing outwards and to compel the expansion movement to go inwards these joints will operate fairly satisfactorily. However, they suffer from other disabilities. They often leak profusely and persistently; they sometimes rust up solid; they only accommodate straight line movement. Of all expansion devices they are probably the least satisfactory, although on very large diameter pipes they are the only type of expansion device that can in many cases be used.

194. THE EXPANSION BELLOWS. The bellows is shown in Fig. 60a. It suffers from some, but not all, of the shortcomings of the sliding joint. It also has some nasty properties that are all its own. As in the sliding joint any pressure inside the bellows tends to blow it apart or rather open it to the fullest extent. The pressure acts on the full area of each fold. If the pipe is properly anchored so that the bellows does not blow out and the expansion movement is compelled to compress the bellows, not only has the anchorage to take the thrust needed to compress the bellows but it must take the blow-open thrust as well. The load on pipe anchorages with bellows-fitted pipes can be very severe indeed. Figures taken in the author's works in 1943 showed that a bellows with four folds for an 8-in. pipe exerted an outward force of $13\frac{1}{2}$ tons with a pressure of 250 psi inside it. An additional force of $5\frac{1}{2}$ tons was needed to compress the bellows

by $\frac{7}{16}$ in.—the amount needed to take up the expansion in 15 ft. of pipe. This is a total load on the pipe anchorages of 19 tons. No wonder that welded-on anchorage lugs get torn off. The pressures and end thrust showed that the effective "pistonlike" diameter of the bellows was just half-way between the pipe diameter and the bellows-fold diameter.

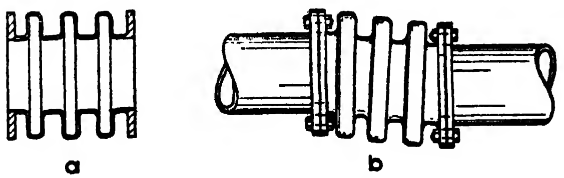


FIG. 60. EXPANSION BELLOWS

The anchorage of pipes fitted with bellows expansions requires great thought and must be very robust, otherwise the bellows will delight in taking up a shape like that in Fig. 60b, with severe local stressing. A bellows with many folds is a wobbly affair. If, therefore, the pipe has a large temperature change and much movement is to be accommodated, it will be necessary to fit a number of sets of bellows. Probably four or five folds are the most that should be used in one set.

A bellows fitted in a horizontal pipe provides many pockets for the collection of condensate. On the other hand, a bellows will accommodate itself to angular and out of line movements in a way that the sliding joint cannot aspire to.

195. THE EXPANSION DIAPHRAGM. The diaphragm shown in Fig. 61 is simply a single "bellow" of large diameter. It is generally used only on very low-pressure apparatus or on plant working under vacuum. With pressures above atmospheric it suffers from a desire to blow apart even more than the

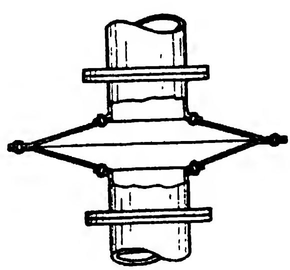


FIG. 61. EXPANSION DIAPHRAGM

bellows owing to the much larger area on which the pressure acts. If used under vacuum conditions the atmospheric pressure tends to close it up, and, if the anchorages are not good, uses up all the movement that should be available for the expansion. It is cheap and simple.

These three devices, the sliding joint, the bellows and the diaphragm all suffer from a serious common disability. The pipe is in effect cut in two and must be held tightly lest it blow apart.

196. THE LYRE LOOP. The Lyre Loop shown in Fig. 62 is an expansion device that is frequently used. It is a strong continuous structure. Pressure inside it cannot blow it apart. There is a slight straightening-out effect, but this is not very great and will not result in forces that cannot be easily taken by reasonably good anchorages. When it expands or contracts the flanges at either end can remain parallel without undue stresses.

The lyre loop is often fitted vertically. If this must be done, owing to lack of space, a drain and trap must be fitted in front of each loop, or trouble with water hammer will occur. Wherever possible it should be fitted horizontally. It is an excellent device, which will accommodate itself to out-of-line and angular movement. It is to be highly commended.

197. THE FULL LOOP. The Full Loop is simply one complete helical turn of the pipe and is shown in Fig. 63. It is simple, robust and occupies less space than the lyre loop. When pressure acts inside it there is a slight tendency to unwind. When it is pulled out cold, or when it is compressed under hot expansion the flanges get out of line and suffer considerable stress, but these stresses seem to have no ill-effects on the joints. The full loop lends itself to horizontal fitting at the end of a horizontal run of pipe. It has a natural fall (unless some bone-head fits a wrong-handed loop), it is more compact than the lyre loop, but theoretically not quite so sound. It is an excellent device.

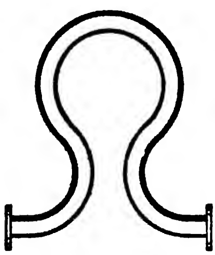


FIG. 62. LYRE EXPANSION LOOP

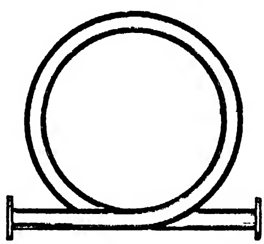


FIG. 63. FULL EXPANSION LOOP

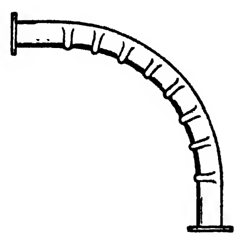


FIG. 64. HALF-CREASED EXPANSION BEND

198. THE CREASED BEND. Fig. 64 shows a half-creased bend. Sometimes lyre loops are made fully creased. Such a bend is more flexible than a plain bend, when it is not under pressure. The half-creased bend is a subtle lop-sided bellows. The outside of the bend is plain, the inside is a bellows. The inside tends to lengthen under steam pressure and hence to straighten the bend. If a half-creased lyre loop is used the whole loop tends to straighten out and put an outward force on the anchorages. Any advantages that creasing may have in increased flexibility may be counteracted by this straightening force that it produces. The original reason for half-creasing bends was to eliminate the thinning that takes place on the outside of a bend during manufacture. This is an advantage, but it possesses the bad qualities of a lop-sided bellows. Fully creased bends and loops should always be installed by specialists.

Loops, bends and lyre loops can be obtained fully creased, when they are simply long bellows. They are extremely flexible, but can move about in all directions and of course are always trying to lengthen.

199. PLENTY OF LENGTH AND BENDS. Given a fair length and plenty of bends a pipe system can accommodate itself very well to expansion. Fig. 65 shows an exceedingly bad arrangement of a branch from a steam main feeding a vertical engine. The tee branch and the engine flange are being cruelly treated. If a loop, such as is shown in Fig. 66, cannot, for some reason, be fitted, a satisfactory result would be got by lengthening the vertical pipes, if this is possible, as shown in Fig. 67. This of course adds to the length of pipe ; increases the heat losing surface ; increases pressure drop and costs more. Unfortunately all the satisfactory expansion devices do all these things.

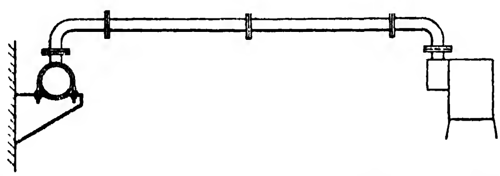


FIG. 65. UNSATISFACTORY PIPING ARRANGEMENT

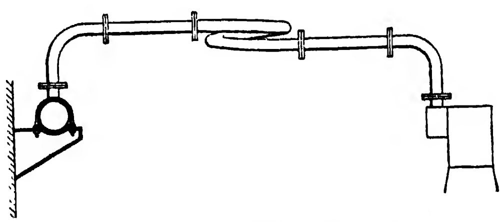


FIG. 66. GOOD PIPING ARRANGEMENT

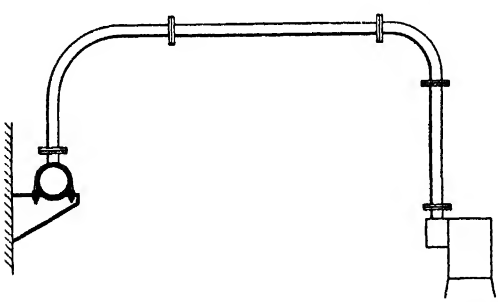


FIG. 67. MODERATELY GOOD PIPING ARRANGEMENT

In Figs. 66 and 67 if it is necessary (as it often indeed is) for the engine valve chest to take the load imposed by pipe expansion, the pipe must be designed to be short, when free and cold, by about half the expected expansion. The engine flange will then be under a " pull " when cold and an equal " push " when hot.

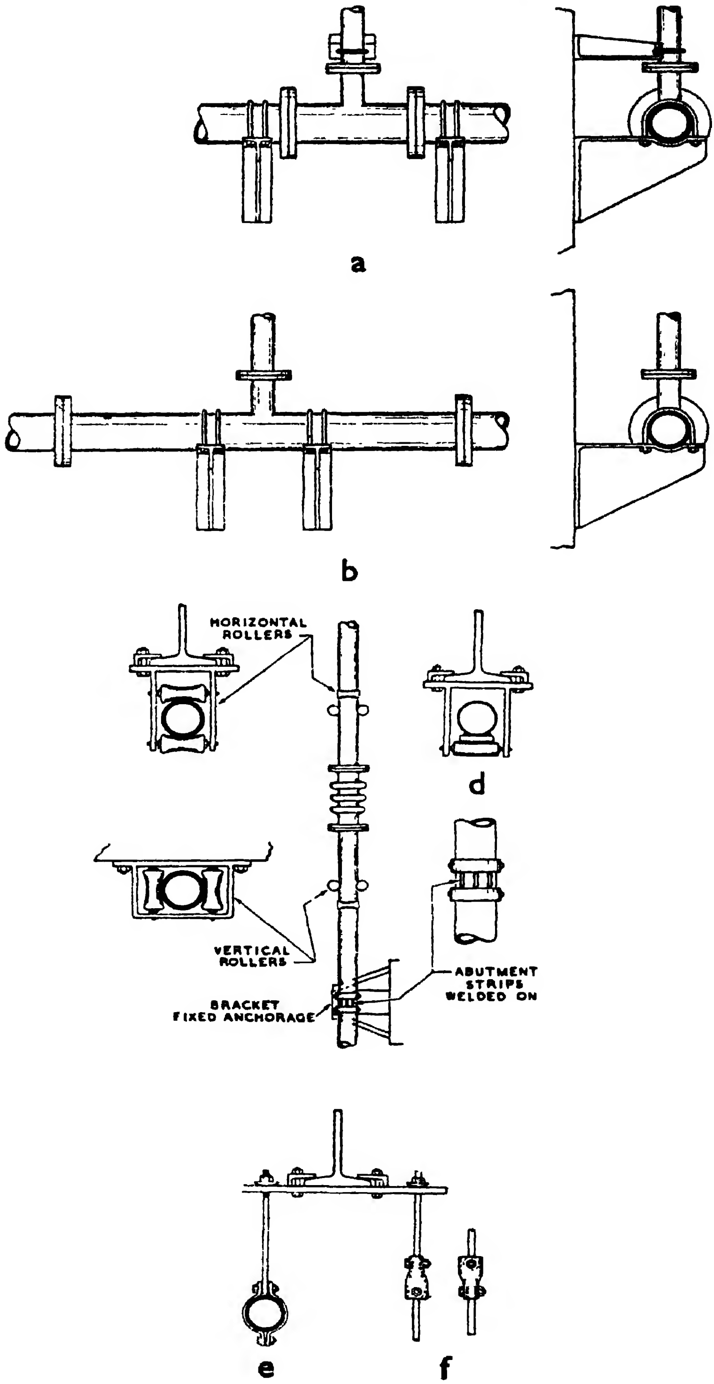


FIG. 68. PIPE ANCHORAGES AND GUIDES

200. ANCHORAGES AND SUPPORTS. The anchoring of steam pipes is very important. Anchorages seldom get the attention they merit. When the amount of expansion that must be taken care of has been worked out, a decision must be made as to where the expansion movement is to be taken up. Fixed definite anchorages must then be placed so as to compel the movement to take place as desired. Fig. 68 shows several forms of anchorages. It is rather difficult to lay down rules.

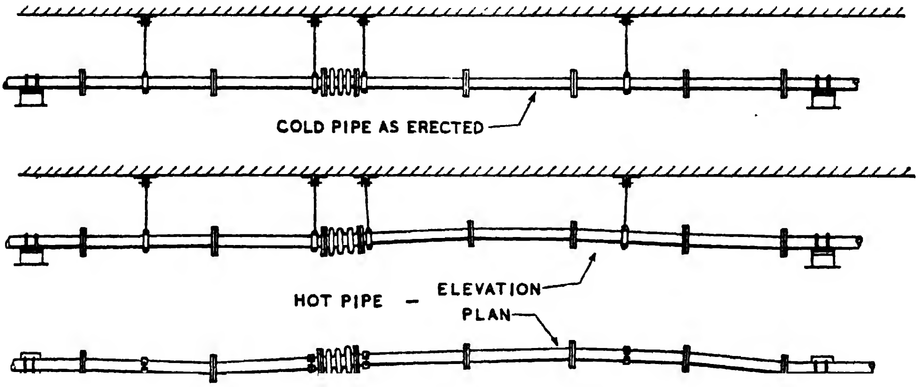


FIG. 69. UNCONTROLLED PIPE EXPANSION MOVEMENT

If bends are to take the movement, the anchorages should if possible be midway between the bends. If loops are provided then the anchorages should be placed at those points where no movement is desired. A tee should almost always be anchored in all directions otherwise the flanges will be racked and stressed. The tee itself should not be anchored, but the pipes immediately adjacent—Fig. 68a.

Spring supports are very common. Their virtue is often largely illusory. When a pipe gets hot the spring may be completely relieved of the pipe load ; or the spring may be fully compressed so that it might just as well not have been provided.

On the other hand proper spring supports are of great value. Where expansion is absorbed by length, bends and loops, lengths of pipe, too long to be unsupported, may move up or down by several inches. In such cases spring supports are essential. There is at least one make of spring support in which, by suitable bell crank leverage, the load on the spring remains virtually constant over a movement of several inches. Spring supports should be fitted by specialists.

Hangers that permit swing should be used with discretion. If hangers support a pipe fitted with bellows, the force needed to compress the bellows is so great that the pipe will probably much prefer to assume a serpentine form. This swings the hangers, causing the pipe's path in the other plane to be a mild switchback in whose dips condensate can collect. See Fig. 69.

Hangers should always be provided with some kind of universal joint. One method is to suspend the hanger on a spherical washer over a clearance hole, Fig. 68e. Another way is to make a simple universal joint out of a short length of tube partly flattened at each end with the flats at right angles, see Fig. 68f.

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Possibly the best form of pipe support between the anchorages is the roller. It can be made to limit movement in any direction, Fig. 68c. It is not much good fitting rollers if they are only used to support the weight of the pipe. Unless they are fitted above as well as below there is nothing to prevent the pipe arching or becoming serpentine. Roller supports necessitate a part of the pipe at the roller being left bare of lagging. Rollers are expensive. Some authorities hold that they stress the pipe unduly due to point contact.

Better contact can be obtained between pipe and roller if a flat pad is welded to the pipe. The pad can then bear on a plain cylindrical roller, Fig. 68d.

Most jobs, especially of small bore piping, do not warrant the expense of roller supports. If a good fall can be given to a pipe a little snakelike movement does no harm.

Pipe-hanging merits more attention than it usually receives. It is very difficult to be categorical and the recommendations given in this Section are by no means of universal application.

The pipes to and from turbines are of great importance. This is especially true of back pressure machines, where the turbine casing is relatively small and may be racked and distorted by the exuberant movement of a huge exhaust pipe. The sizing, hanging and expansion arrangements of pipes to and from a turbine should always be the responsibility of the turbine builder.

201. THE MYSTERIES OF EXPANSION. In many factories no provision is made for expansion and they get away with it. Expansion movements can be very queer. In the author's factory there was for many years a large steel mixer 60 ft. long and 10 ft in diameter. This was rigidly tied to a dozen cast iron columns in an old brick building. Every week-end there must have been a movement of nearly ¾ in., but it was never detected ; the brick walls were not pushed out ; the cast iron columns were not cracked. What is the answer ? It is not " Don't let's bother. It will be all right ". It is " Take all reasonable precautions and if they fall short of perfection it may be all right ".

Piping, if working at fairly high temperature, should be designed as if it were half to three-fifths hot. This will call for a lot of pulling up during cold erection. Actually it means that if an expansion movement of 5 in. is to be expected in a piping system, the cold pipe must be pulled up 2½ in. to 3 in. during cold erection. One of the advantages of expansion loops is that they move in either direction with almost equal readiness. Expansion loops should be designed so that when cold, half to three-fifths of the total expected movement must be pulled up.

The amount of movement that expansion loops can accommodate cannot be given here. It depends on the thickness of the pipe wall as well as on the pipe diameter and the working temperature. The movement and the load imposed on the anchorages should be obtained from the makers. For rough guidance it is generally safe to say that a standard expansion loop will accommodate a movement about equal to half the bore of the pipe.

202. FEEDING BRANCHES. When steam flows past a branch it has momentum and must be forced to turn down the branch by means of a pressure drop. On the upstream side of the branch the velocity is higher than on the

downstream side. The steam acquires this velocity energy by stealing some of its own pressure energy. On the downstream side of the branch the steam flows slower and the velocity head is converted back into pressure head so that there is a greater pressure beyond the branch than above it. If the straight pipe and the branch feed identical plant the branch tends to be starved.

Fig. 70 shows a simple compact pipe arrangement for feeding the four steam inlets of a large evaporator. Inlet A is nearest the steam supply ; inlet C is furthest from the supply. It might be expected that inlet A would get the

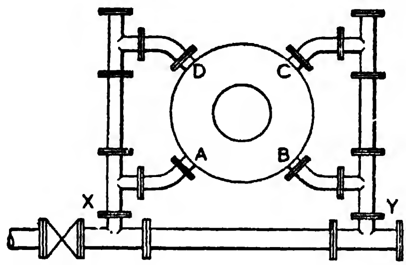


FIG. 70. SIMPLE ARRANGEMENT OF PIPING WILL STARVE BRANCH A AND GIVE BRANCH C EXCESS

most steam and C the least. This will probably not be so except at very low steam flows when the steam velocity is so small that there is little momentum and little velocity head. At high steam flows branch X may be starved to the benefit of branch Y and branches A and B may be starved to the benefit of D and C. So that inlet A may get least steam and inlet C may get most. Unequal steam supply between the inlets may upset the circulation in the evaporator and should be avoided.

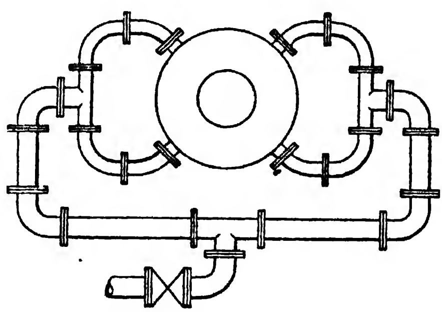


FIG. 71. ARRANGEMENT OF PIPING TO GIVE PERFECTLY EVEN STEAM DISTRIBUTION

Fig. 71 shows the correct method of feeding the plant, using standard fittings. The steam to each inlet goes an identical distance through identically shaped piping. The number of Tees that the steam has to turn through are the same as in Fig. 70, but each path has three extra bends. This is the forfeit that must be paid for equal distribution. In addition to even distribution, the arrange-

ment in Fig. 71 is very flexible and will deal with its own expansion movements, whereas the arrangement in Fig. 70 is rigid and calls for very careful design, workmanship and erection. The arrangement shown in Fig. 71 is all on one plane. Greater compactness may be obtained by putting some of the system in the vertical plane. Lower resistance can be got by using Y pipes instead of Ts. Fig. 72 shows what might be called the ideal arrangement.

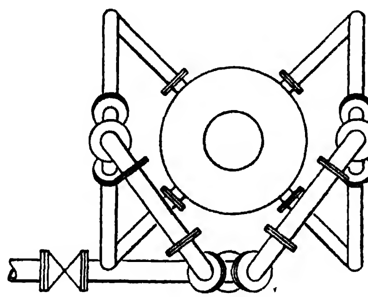


FIG. 72. IMPROVED ARRANGEMENT OF PIPING TO GIVE EVEN STEAM DISTRIBUTION WITH REDUCED OBSTRUCTION TO STEAM FLOW AND FLEXIBILITY FOR EXPANSION

The Institution of Mechanical Engineers have a section of old water main which was part of London's water network some hundreds of years ago. In those days they hollowed tree trunks to make pipes and thus had to make virtues of necessities when forming a branch. They had to choose a tree that had a natural easy Y shape. The coming of cheap castings has made our piping systems much more resistant to flow than they might be.

203. TYPES OF PIPING. There are four broad types of steam piping systems in use :—

(*a*) Flanged cast iron.
(*b*) Screwed steel.
(*c*) Flanged steel.
(*d*) Welded steel.

Cast iron flanged piping is very easy to erect. It will hold itself straight between hangers. It does not corrode or rust. It lends itself to easy alterations and additions. Awkward bends can be cheaply made. It is heavy. It has a large number of flanges that must be lagged at extra expense. Expansion movement incorrectly controlled is liable to crack flanges.

Each type of steel piping has certain advantages and disadvantages over cast iron. Steel is much lighter. Flanges can be much fewer. It will accommodate uncontrolled expansion movement fairly readily. It has a smaller surface to lose heat. It requires more frequent and careful hanging in order to prevent sagging.

Screwed steel piping is cheap. It is easy to erect. Flanges are almost eliminated. Complicated arrangements require many unions to enable them to be erected. It does not lend itself to easy additions or alterations. Any leaks

that appear may be very difficult to cure as most joints cannot be tightened without disturbing the joints on either side.

Flanged steel piping combines some of the advantages and disadvantages of flanged cast iron and any form of steel piping. The pipe lengths can be much longer than cast-iron lengths, so that there can be relatively few flanges.

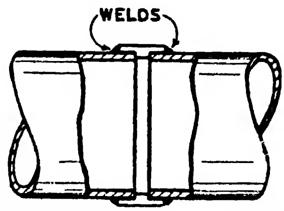


FIG. 73. WELDING SOCKET

Welded steel piping is probably the ideal for most process liquors, for water and for low-pressure steam. High pressure piping requires the experience of specialists. The very ease with which welded piping can be made of any shape is a danger unless the utmost care is used and meticulous testing and inspection carried out. Flanges can be reduced to a minimum, but erection should not be carried out by welding. Welds should, whenever possible, be done downwards as upward welding is difficult and uncertain. Welding sockets, Fig. 73, greatly facilitate erection of water and liquor pipes. The pipes can be offered up in the erected position, the sockets tack-welded and the assembly lowered. The maximum length that can be conveniently handled can then be welded up accessibly, flanges fitted and the length tested. Without sockets much time is wasted in offering the ends and the tack welds have to be much more elaborate. The welding socket doubles the length of the weld, but more than halves the time of erection. The welding socket may prove unsatisfactory on some lines. There are two crannies at each joint where corrosion can start. If the material to be piped is known to be liable to attack steel pipes it will be better to make the weld without any internal or external socket. Sockets are not usually advisable for steam pipes. Without sockets clamps of various sizes and kinds must be used while making the tack welds with the pipes offered together and to hold the pipes correctly when being lowered lest the tacks be broken.

Welding seems so easy and quick that the greatest restraint and caution must be used lest risky practices develop.

Specialists should always be employed for high pressure piping.

Cast iron piping should not be used for high pressure steam, nor for steam with much superheat. It is quite satisfactory for steam at pressures below 100 psi, but the expansion arrangements must be carefully watched. The only expansion joint possible with cast iron is the sliding expansion joint. Cast iron must be used for the vapour pipes of many evaporators due to the corrosive nature of the vapour from many materials.

Whatever kind and type of piping is used wherever flanges occur, they should be made and drilled to British Standard Specification. Table XVIII gives some figures covering the more common sizes.

204. THE COST OF PIPING. When considering at what pressure steam should be distributed, the relative costs of large and small piping is constantly being considered. The cost must vary from job to job. Awkward situations and inaccessible places will increase the relative cost of the larger sizes. Many bends and tees also increase the cost but have little effect on the relative costs between the sizes. Table XXXI gives costs of erecting and lagging steel piping in the author's factory in 1957 in accessible positions.

TABLE XXXI. APPROXIMATE COST OF BUYING, ERECTING, LAGGING AND SHEETING 100 FT. OF STRAIGHT STEEL PIPING WITH WELDED JOINTS (1957)

NOMINAL PIPE DIAMETER	COST OF 100 FT. OF ERECTED PIPE		LAGGING 1½″ THICK AND SHEETING		TOTAL
	PIPE	LABOUR	MATERIAL	LABOUR	
INCHES	£ s. d.	£ s. d.	£ s. d.	£ s. d.	£ s. d.
2	11 18 5	8 17 9	28 7 0	10 2 0	59 5 2
3	17 8 5	13 6 6	33 18 2	12 12 6	77 15 7
4	31 3 10	16 15 9	39 2 8	15 3 0	102 5 3
5	37 0 2	22 14 9	44 1 8	19 4 3	123 0 10
6	52 19 4	25 13 6	49 4 5	23 5 6	151 2 9
7	68 15 3	33 11 6	54 5 0	28 6 6	184 18 3
8	89 12 0	39 10 0	59 14 4	32 17 9	221 14 1
9	97 10 4	45 18 5	64 16 2	36 9 0	244 13 11
10	112 2 1	51 7 0	69 16 0	40 10 3	273 15 4

Easy figures to remember are (at the time of writing). 5s. a foot for 2 in. piping, 10s. a foot for 5 in., 15s. a foot for 8 in. pipe and £1 a foot for 10 in. piping lagged and sheeted.

205. VALVES AND COCKS. Valves always give some trouble, always require quite a lot of maintenance and often leak. It generally pays to buy an expensive, high-class valve. For steam the parallel slide valve is probably the best, but it should not be used in the cracked open position (nor should any steam valve for that matter except those special valves designed specifically for accurate control at very small openings) and it is very difficult to train operators not to screw the spindle of a parallel slide valve down tight. The valve wheel should be turned to the limit of its travel and then eased back at least half a turn. If the valve is screwed hard down on to the body there is every chance that the loose joint faces may get askew and leak. The loose joint faces should be free to take up a flat position against the seating.

For process liquids, including water, the rubber diaphragm valve is outstanding. It has no gland, it does not object to dirty gritty liquids, but it has several limitations. Most operators close it much too tight and damage the diaphragm. It is not always satisfactory on hot liquids and it cannot be used under vacuum.

Cocks are tempting. Their position can be seen at a glance. Three-way arrangements are easily made. None the less, the ordinary metal to metal cock should be avoided like the plague. It leaks profusely, it wears quickly, it quickly jams solid and its upkeep is tremendous. Some lubricated cocks are, however, quite satisfactory in applications where lubricant can be tolerated. There is also at least one make of packed cock which gives good service.

Valve sizes are most important. From Section 174 it might be thought that the valves should be as large as possible to reduce the resistance to steam flow. This is quite true of a simple stop valve which is either full open or tight shut. If the valve is used for controlling the steam flow to a piece of plant, whether automatically or by hand, the valve size is settled by control considerations.

Consider an ordinary globe valve. When the valve is well clear of the seat the amount of steam that will flow depends on the area of the circular orifice of the valve seat and the pressure drop across it. When the steam has passed through the seat orifice it must then turn at right angles and pass through the circular slot between the seat and the valve. Let us see how high the valve must be lifted off the seat to provide an annular slot equal in area to the valve seat orifice.

If D is the diameter of the seat orifice, the area of the orifice is $\frac{\pi D^2}{4}$.

The area of the annular slot is :—

$$\text{Circumference} \times \text{Lift} = \pi D \times \text{Lift}.$$

If the slot area and the orifice area are to be equal, we have :—

$$\pi D \times \text{Lift} = \frac{\pi D^2}{4}$$

$$\therefore \quad \text{Lift} = \frac{\pi D^2}{4\pi D} = \frac{D}{4}.$$

If a 2 in. valve, full open, will pass all the steam required, the lift will be $\frac{2}{4} = \frac{1}{2}$ in. off the seat.

Now the pressure drop across this full open valve is very small. Suppose it is desired to cut the heat flow to a heating surface in half. The temperature difference between the steam and the material—say boiling water—must be halved. If, when the valve was full open, the steam pressure was 30 psi.g., the temperature difference will have been $274 - 212 = 62°$ F. The reduced temperature difference must be about 31° F., so that the reduced steam temperature must be $212 + 31 = 243°$ F. corresponding to steam at 11·5 psi.g. The pressure drop across the valve will now be 18·5 psi where before, when the valve was full open, it was perhaps ·25 psi. Now the flow varies approximately as the square root of the pressure drop. So that the flow with 18·5 psi pressure drop will be :—

$\sqrt{\frac{18\cdot 5}{\cdot 25}} = \sqrt{74} = 8\cdot 6$ times as much as with the full open valve with a pressure drop of ·25 psi.

The valve must therefore be only opened $\frac{\cdot 5}{8\cdot 6} = \cdot 058$ in.

Now suppose a 4 in. valve had been used in place of a 2 in. The annular orifice would be double that of the 2 in. valve and the valve would only have to be opened about ·03 in. Regulation at this minute lift would be almost impossible by hand, and in an automatic reducing valve would certainly result in hunting. The steam would all be blowing across the seat which would be quickly scored and damaged.

It may therefore often be wise and perfectly legitimate to use a 4 in. pipe terminating in a 2 in. valve.

The diameter of the pipe is determined by the permissible pressure drop. The diameter of the valve is determined by the steam flow that the valve is to control.

Many a reducing valve has been cursed and thrown out simply because it was too big.

Where, in spite of correct sizing, large flows must be combined with very small, accurately controlled flows, it is essential to use a large valve, byepassed by a small valve. In this way accurate control can be secured over a very wide flow range without damage to the valve seatings. Some automatic valves have their openings so shaped as to give accurate control over very small flows while permitting very large flows.

* * *

CHAPTER 6

INSTRUMENTS — MEASUREMENT

> He made an instrument to know
> If the moon shines at full or no ;
> . . . .
> For he, by geometric scale,
> Could take the size of pots of ale ;
> Resolve by sines and tangents, straight,
> If bread and butter wanted weight ;
> And wisely tell what hour o' th' day
> The clock does strike, by algebra.
>
> SAMUEL BUTLER. *Hudibras.* 1663.

MEASUREMENT is the essence of all factory economy. We must know the pressure and temperature of steam in all steam-using plant. We ought also to know the quality and the quantity of the steam. Instruments are available to indicate and record all these things except quality. Quality can only be determined by a somewhat tiresome test ; there is no proper instrument which will indicate or record the wetness in steam. Measuring instruments are useless if they tell lies. It never pays therefore to buy cheap instruments.

206. BOURDON PRESSURE GAUGES. The Bourdon gauge has been briefly described in Section 27. It is moderately robust but not always accurate. Bourdon gauges need regular checking. They are fairly consistent, so that, once their error is known, the corrected gauge reading can be relied on for some time.

The Bourdon pressure gauge consists of a very much flattened oval-sectioned thin tube bent into an almost complete circle, shown edgewise in Fig. 74. The pressure to be measured is led into the inside of the curved flattened " Bourdon " tube which is closed at its other end. The open end of the Bourdon tube is fixed to the frame of the gauge. The closed end of the Bourdon tube is free to move and is attached by a short link to the short arm of a quadrant. The quadrant is pivoted on the frame of the gauge and its teeth mesh with a pinion on the pointer axle.

It will be seen that the Bourdon tube is almost flat. When pressure acts on the inside of the tube it tends to blow the tube from a flat into a rounded section. This forces the tube to bend outwards so as to form part of a larger circle. The lengths of the inner and outer peripheries of the tube cannot change. If these two peripheries are blown further apart they must take up a form as part of larger diameter circles. If the distance between the inner and outer walls of the tube increases by 10 per cent. the diameter of the circle to which the tube is bent will increase by about 10 per cent.

In Fig. 74 the Bourdon tube shown is bent into a circle about $1\frac{1}{4}$ in. diameter, and the walls are about ·05 in. apart. If the pressure is sufficient to blow the walls ·003 in. further apart it will cause the tube to unbend into a circle six per cent. larger in diameter. This new circle is shown dotted in the diagram and the free end will move so as to pull the quadrant sufficient to turn the pointer right round the scale on the dial.

The pressure inside the Bourdon tube is opposed by the natural spring of the tube and the pressure of the atmosphere.

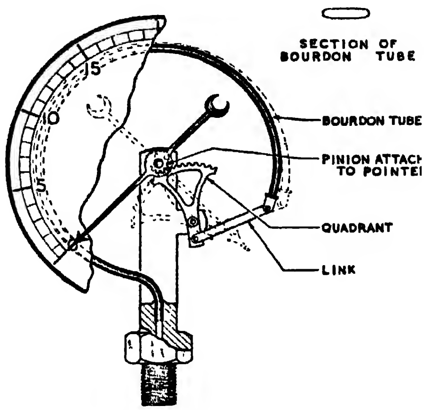


FIG. 74. BOURDON PRESSURE GAUGE

The readings of Bourdon gauges can be checked over low ranges either by means of a water column or a mercury column. If the factory has a hydrant system supplied from a high level tank, connections can be made at various heights in some hydrant pipe. Provided there is no flow in the pipe, the pressure recorded is exactly the equivalent of the measured hydrostatic head. This is a very accurate and convenient method. Pressure equivalents are given in Table XXXII, p. 168.

If there is no high water tank, gauges can be compared with a standard pressure gauge kept specially for testing and used for no other purpose. This check gauge should be sent to the makers periodically for recalibrating—and the makers should be told what the gauge is to be used for.

If there are many gauges in the factory, with many scale ranges it may be worth while getting a dead weight gauge testing machine.

Table XXXII can be used to get any figures thus. Suppose the mercury and water equivalents to 98·4 psi are wanted :—

Psi		*In. Hg.*		*Ft. H_2O*
90		183·24		207·81
8		16·288		18·472
·4		·8144		·9236
98·4	=	200·3424	=	227·2056

207. MERCURY PRESSURE GAUGES. The mercury gauge is more suitable than the Bourdon gauge for low pressures and for pressures below atmospheric pressure. The mercury gauge has been briefly described in Section 28.

Fig. 75 shows a glass U-tube part filled with mercury. If one limb of the tube is connected to a vessel under a very low pressure or to a vessel under vacuum, the difference in height of the mercury in the two limbs shows the pressure difference, plus or minus, between the atmospheric pressure, which is pressing on the open end, and the pressure in the vessel.

Open U-tubes such as this indicate "gauge pressure" or "apparent vacuum". In order to find the true or absolute pressure a correction must be made for the height of the barometer.

Fig. 76 shows a glass U-tube with one end closed. If the tube is filled with mercury and then tilted so as to remove completely all air from the left hand or

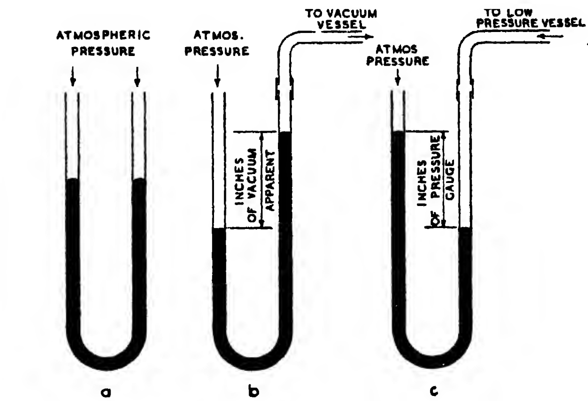


FIG. 75. ATMOSPHERIC MERCURY U-TUBE

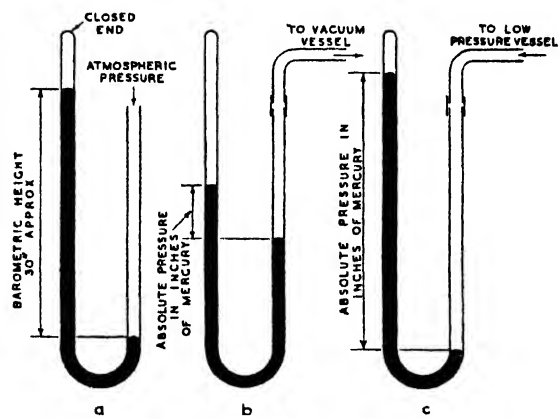


FIG. 76. ABSOLUTE PRESSURE MERCURY U-TUBE

closed limb there will be no pressure acting on this limb. The difference in height of the mercury in the two limbs, if the open end is connected to a vacuum or low pressure vessel, will then show the absolute pressure or the true vacuum.

The readings in inches of mercury can be readily converted to psi by means of Table XXXII.

208. WATER GAUGE. For very slight pressures, such as those produced on the suction or delivery sides of fans, the mercury column does not give a sufficiently sensitive reading. Quite a good draught from a fan would only register about ⅛ in. on a mercury column. The U-tube can be filled with water instead of mercury when the readings will at once be multiplied by nearly 14. Only the open tube, Fig. 75, can be used with water. The closed tube shown in Fig. 76 would have to be over 30 ft. high were water to be used. It is however seldom or never that very small absolute pressures must be measured. Generally such measurements are for finding the suction or pressure of a fan which means the difference between atmospheric pressure and the pressure in the air duct, or the difference in pressure across some resistance such as a grate. The draught or pressure difference is then measured in " inches of water ".

TABLE XXXII. PRESSURE EQUIVALENTS

PSI	INCH HG.	FOOT H_2O
·4332	·8819	1
·4912	1	1·134
·8664	1·7638	2
·9824	2	2·268
1	2·036	2·309
1·2996	2·646	3
1·4736	3	3·402
1·7328	3·528	4
1·9648	4	4·536
2	4·072	4·618
2·1660	4·4095	5
2·4560	5	5·670
2·5992	5·2914	6
2·9472	6	6·804
3	6 108	6·927
3·0324	6·1733	7
3·4384	7	7·938
3·4656	7·0552	8
3·8988	7·9371	9
3·9296	8	9·072
4	8·144	9·236
4·4208	9	10·206
5	10·180	11·545
6	12·216	13·854
7	14·252	16·163
8	16·288	18·472
9	18·324	20·781

209. MERCURY IN GLASS THERMOMETERS. Mercury in glass thermometers have been briefly mentioned in Section 29. These thermometers are the type in most general use and they work because mercury expands a

great deal with heat, whereas glass only expands a little. The thermometer consists of a very narrow glass tube with a bulb at the bottom. The bulb is filled with mercury and the air is pumped out of the tube above the mercury and the tube is then sealed up. When the thermometer is placed in something hot both glass and mercury get heated. The mercury expands a great deal but the glass only expands a little. The glass does not expand nearly enough to make room for the hot mercury in the space it previously occupied. The mercury therefore pushes its way up the tube and the distance it has pushed itself up is a measure of the temperature. While mercury in glass thermometers are made to register temperatures of 900° F. or so, it is not recommended that they be used for temperatures above 500° F.

210. MERCURY IN STEEL THERMOMETERS. Another type of thermometer in common use in industry is the mercury in steel type. This consists of a steel bulb connected by a very fine steel tube to a Bourdon pressure gauge. Tube and bulb are filled with mercury. When the bulb is heated the mercury expands more than the steel and as the mercury cannot escape it can only occupy the necessary extra space by blowing out the Bourdon tube of the pressure gauge whose dial can be marked in temperature degrees instead of pressure units.

This type of thermometer is particularly useful where it is desired to read the thermometer some distance from the place where the temperature is being measured

211. PYROMETERS. For very high temperatures mercury cannot be used. High temperature thermometers—usually called "Pyrometers" ("fire meters")—are usually electrical. There are two common types, resistance and thermocouple. Resistance pyrometers consist of a coil of wire protected by a covering. The coil is placed in the vessel, flue or pipe whose temperature is required. An electric current is passed through the coil and the current that can pass diminishes as the temperature rises. An electric measuring instrument is put in the circuit and the amount of current flowing can be indicated. This current flow is a measure of the temperature and the instrument can be calibrated in temperature degrees instead of in electrical units.

The other type depends on the curious property possessed by certain metals. If two strips of certain different metals are joined together in perfect contact at one end, an electric current will flow through a circuit attached to the unjoined ends if the metal junction is heated. The hotter the metal junction the greater the current. An electrical instrument placed in the circuit can be calibrated to read temperature.

The junction of the two metals is called a "Thermocouple". Thermocouples have a very large field of use. Minute couples can be peened into the actual wall of a boiler tube, evaporator or heat exchanger. Not only can the thermocouple be put anywhere, but it responds almost instantaneously to temperature change. By means of D.C. amplifiers (wireless valves) minute currents can be magnified to operate distant large instruments or to work automatic temperature controls.

212. FAHRENHEIT AND CENTIGRADE. The origin of these two temperature scales is described in Sections 30 and 31. Although in this book only the Fahrenheit scale is used the author feels that it is important that the Centigrade scale should be generally adopted, especially for all steam uses. With 0 as the freezing point of water the sensible heat of water at low temperatures is the same on the Centigrade system as the temperature. There is no need to remember to add or deduct 32. This removes one fruitful source of error and greatly facilitates mental calculations.

As $-40°$ F. and $-40°$ C. are the same temperature on either scale, a very simple conversion can be done from one scale to the other thus : Add 40 to the known temperature. To find Centigrade from Fahrenheit take five-ninths of this figure. To find Fahrenheit from Centigrade take nine-fifths of the figure. Then, in either conversion, deduct 40.

213. MEASURING STEAM QUALITY. Measurement of wetness or superheat, steam quality, should be done far oftener than it is. The quantity of condensate from a piece of plant is often measured and it is generally assumed that the weight of condensate, perhaps corrected for flash, perhaps not, is the weight of steam that was used. More often than not this is quite wrong. The measurement of quality is quite simple, but it takes a little time and trouble and the result of a single measurement cannot be trusted. The real difficulty is to ensure that the sample of steam tested is representative. At least six measurements should be taken and averaged, and the result must only be looked upon as a good indication, not an exact measurement.

The apparatus for measuring steam quality is called a calorimeter (or heat measurer) and there are three types. The first is the separating calorimeter which is alleged to remove the water droplets from the steam by a series of changes of direction. If, as is often the case, the water is present as a fine mist, separation is very incomplete and the results are not trustworthy. If this calorimeter were really reliable all our difficulties with steam driers would disappear. We should just fit large separating calorimeters.

The second is the throttling calorimeter. This works by wiredrawing the steam almost to atmospheric pressure. From the temperatures and pressures before and after the expansion the wetness may be calculated. This has two disadvantages. The superheating effect is very small at low pressures and can only evaporate a small amount of wetness. There is no guarantee that all the moisture has been evaporated. The throttling calorimeter has the advantage that it can be fitted to a pipe and left there, and readings can be quickly and easily taken as often as desired. Provided the wetness is very small this may be very useful, as a watch can be kept on quality.

The third type is the bucket calorimeter in which steam is condensed in a measured amount of cold water. The increase in water weight and the rise in water temperature give the quality of the steam. As a rule neither of the first types is used alone. Only the bucket type will be described here, firstly, because for all-round use it is the best; secondly, because it requires no special apparatus.

Whenever possible the sampling point should be arranged in a vertical length of pipe, where it is more likely to give a true sample. But if the steam pipe, from which the sample steam is to be drawn, is horizontal, fit a

connection with a valve or cock into the *top* of the pipe. The connection should be very small if the method now to be described is used. It should be a pipe not more than $\frac{3}{8}$ in. bore, preferably less. It should project right across the pipe that is being sampled, should have its end plugged, and should have a number of small holes about $\frac{1}{16}$ in. drilled on its upstream side, see Fig. 77. The sampling pipe should be well lagged.

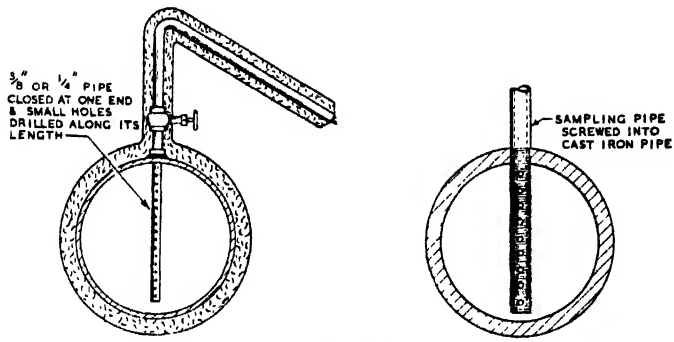


FIG. 77. STEAM SAMPLING PIPE

Fit a pressure gauge, whose accuracy has been checked, in the pipe close to the sampling connection. Take a thermos flask and bore three holes in the cork. Through one hole insert a thermometer extending about two-thirds down the flask Through another hole insert a glass or metal tube extending about three-quarters into the flask. The third hole is simply an expansion vent.

Weigh the flask, cork, thermometer and tube, w_1.

Fill the flask two-thirds full of cold water.

Weigh again, w_2.

Then $w_2 - w_1 = w_3$, the weight of the water.

Shake the flask well.

Read the temperature of the water t_1.

Then $(t_1 - 32)w_3 = h_1$, the total heat in the cold water in Btu.

Blow steam through the sampling connection to warm it. Attach a short rubber tube from the pipe sampling connection to the thermos tube.

Open the sampling valve cautiously and allow steam to bubble into the water. Shake the flask gently and continuously and watch the thermometer. Take the greatest care that all the steam is condensing. While the steam is bubbling in, read the steam pressure in the pipe. When the temperature has risen to about 150° F. or 170° F., partly close the valve, remove the rubber tube, shake the flask well and read the temperature t_2.

(The reason for only partly closing the valve is to make quite certain that none of the water in the flask is drawn out due to a partial vacuum forming in the sampling tube by condensation.)

Weigh the flask again, w_4.

Then $w_4 - w_1 = w_5$, the total weight of the water and the condensed sample of steam.

The total heat after condensation is $w_5 (t_2 - 32) = h_2$ Btu.

The weight of the sample is $w_5 - w_3 = W$.

The total heat in the sample is $h_2 - h_1 = H$.

The total heat per pound of steam as sampled will be $\frac{H}{W}$. Find the heats of saturated steam at the sampling pressure in the steam table. Call the total heat Q, the sensible heat S and the latent heat L.

If $\frac{H}{W} = Q$ the steam was dry saturated.

If $\frac{H}{W}$ is less than Q the steam was wet.

If $\frac{H}{W}$ is more than Q the steam was superheated.

WET STEAM.

If the sensible heat S is deducted from $\frac{H}{W}$ the remaining heat must have been latent heat.

This remainder divided by L will give the dryness fraction.

$$\text{So} \quad \frac{\left(\frac{H}{W} - S\right) 100}{L} = \text{per cent. dryness.}$$

SUPERHEATED STEAM.

$$\frac{H}{W} - Q \text{ must equal the superheat in Btu,}$$

$$\text{and } 2\left(\frac{H}{W} - Q\right) \text{ is roughly equal to the superheat in } {}^\circ\text{F.}$$

EXAMPLE. A Ruths accumulator supplies steam to a pair ot mains at 15 psi.g. Steam samples were taken in each pipe about 150 ft. from the accumulator.

The measurements are set out on the opposite page.

The effect of errors will be as follows :—

	Pipe A.	*Pipe B.*
If the weight of the sample had been underestimated by 1 gram (·0022 lb.) a ·14 per cent. weighing error, the dryness would have been	87·5 per cent.	87·3 per cent.
If the temperature of the cold water had been underestimated by 1° F., a 1·4 per cent. error, the dryness would have been	91·8 per cent.	90·6 per cent.

	Pipe A.	*Pipe B.*
If the temperature of the hot water had been underestimated by 1° F., an error of ·7 per cent., the dryness would have been	95·1 per cent.	92·9 per cent.

Measurements:—

Weight of flask, tube and thermometer, w_1	·8102 lb.	·8102 lb.
Weight of flask, etc., and cold water, w_2	1·4330 lb.	1·3849 lb.
Weight of cold water, $w_3 = w_2 - w_1$ ..	·6228 lb.	·5747 lb.
Temperature of cold water, t_1 ..	70° F.	74° F.
Heat in cold water, $w_3 (t_1 - 32) = h_1$ ..	23·666 Btu	24·137 Btu
Weight of flask and hot water, w_4 ..	1·4742 lb.	1·4398 lb.
Weight of hot water, $w_5 = w_4 - w_1$ ..	·6640 lb.	·6296 lb.
Temperature of hot water, t_2 ..	136° F.	165° F.
Heat in hot water, $w_5 (t_2 - 32) = h_2$ ..	69·056 Btu	83·737 Btu
Weight of sample of steam, $w_5 - w_3 = W$	·0412 lb.	·0549 lb.
Heat in sample, $h_2 - h_1 = H$	45·390 Btu	59·600 Btu
Total heat per lb. of sample $\frac{H}{W}$	1101·7 Btu/lb.	1084·9 Btu/lb.
Steam at 15 psi.g. has total heat Q ..	1,164·5	1,164·5
Steam at 15 psi.g. has sensible heat S ..	218·4	218·4
Steam at 15 psi.g. has latent heat L ..	946·1	946·1
$\frac{(\frac{H}{W} - S)\,100}{L}$ = per cent. dryness ..	93·4 per cent.	91·6 per cent.

We see that by far the most important measurement is the weight. In the example given the flask was weighed on a rough chemical balance and the accuracy was probably within ·1 gram. Accuracy within ¼ gram is desirable but the difficulty of being certain that the steam sample is a fair average is almost unsurmountable.

If fairly accurate weighing apparatus is not available a rough result can be obtained by measuring the water in a graduated glass cylinder.

Where a small accurate balance is not available and greater accuracy is desired than can be obtained by measurement the thermos flask can be replaced by a barrel containing 100 to 200 lb. water. The barrel should be of wood to reduce radiation loss and can rest on the platform of a potato scale. The water must be well mixed up to get fair temperature readings. In both thermos or barrel the cold water should be as cold as possible as this gives a much bigger heat capacity so that weighing errors will not have so great an effect. In the example given the water used was drawn from the cold water service in a warm department. Any measurements must be repeated a number of times and average results taken.

If a barrel is used instead of a vacuum flask an allowance should be made for the heat capacity of the material of the barrel. If the barrel is made of steel the whole of its heat capacity should be taken into account. If the barrel is of wood it will not heat right through owing to its low heat conductivity. The method of making the correction is : The specific heat of steel is ·15 and of oak ·57.

To the heat in the cold water, h_1, add the heat in the cold barrel :—

Weight of steel barrel $\times$ ·15 $\times$ $(t_1 - 32)$

or Weight of oak barrel $\times$ ·57 $\times$ $(t_1 - 32)$.

To the heat in the hot water, h_2, add the heat in the hot barrel: —

Weight of steel barrel $\times$ ·15 $\times$ $(t_2 - 32)$

or Weight of oak barrel $\times$ ·57 $\times$ $\left(\frac{t_2 + t_b}{2} - 32\right)$

where t_b is the outside temperature of the barrel. This is very difficult to measure and must probably be estimated by touch.

If very cold water, say iced, is used, it may be found that the steam must be bubbled in very slowly or it will fail to condense. It is a curious thing that very cold water is said to condense steam more slowly than warmer water. This is possibly due to the higher viscosity of icy water.

A large error can occur if the steam is bubbled into the condensing water so fast that it does not all condense. All the moisture in the steam that passes into the flask is collected but only part of the dry steam is condensed. For example, suppose the steam is really 20 per cent. wet. If it is bubbled in so fast that only half of it condenses only half the latent heat in the sample will have been secured but the whole of the moisture will have been collected. The test will therefore say that the steam was about 40 per cent. wet.

214. SAMPLING WET STEAM. The difficulty of drawing off a truly representative sample of wet steam from a pipe is great. When steam carrying water droplets is flowing along a pipe it is only possible to get a true sample if the sample is drawn off at the same speed as the steam is flowing along the pipe. If the sample is drawn off slower than the steam flow there will be too much moisture in the sample because the momentum of the water droplets will cause them to fly into the sampling pipe while the dry steam is deflected round. If the sample is drawn off faster than the steam flow the sample will be too dry because the true vapour will be drawn into the sampling pipe but much of the fair share of this vapour's moisture will go past the sample point, because of the momentum of the heavier droplets. It is useful to try to get some idea as to whether this is in fact giving unreliable figures by taking some of the samples as fast as condensation will permit, and taking some very much more slowly. In general, it can be said that using a barrel and a small pipe—say ½-in.—the sample will be drawn off too fast, whereas with the thermos flask the sample will be drawn off too slowly however small the sampling tube.

215. MEASURING STEAM FLOW. The best way to measure the flow of steam is by means of a proper steam meter. These are high grade instruments, require skilled maintenance and are expensive. In some operations such as the

heating of a quantity of process material with open steam, where the steam input is so great that the loss by radiation must be of small importance, it is possible and sometimes legitimate to take the steam consumption as being, say 5 per cent. more than the theoretical. In other cases it may be possible to measure the condensate. This method is only accurate if the necessary corrections are made for flash and for the quality of the input steam. The commercial steam meters will not be described; their description is available in their makers' literature. Condensate measurement and calculation will be dealt with in Sections 234 and 235. A cheap simple steam meter that any handy man can make out of standard pipe fittings will be described in detail. It can be used with complete trust in the readings obtained.

216. COST OF HOME-MADE STEAM METER. The meter to be described is very cheap, but then it only indicates; it does not record; it does not integrate. These are the very valuable things that we pay for when we buy a good commercial meter. It is, however, a valuable instrument. If is it moved from pipe to pipe every week or so it will probably show up such interesting points that a desire for more ambitious recording instruments will be awakened.

The cost of making one of these meters in the author's factory in 1943 was :—

	£	s.	d.
Pipe fittings	4	3	7
Material for scale and protector		6	0
Labour—			
Apprentice		19	0
Carpenter		8	2
	5	16	9
Mercury	2	15	0
	£8	11	9

If the making of the meter seems too much of a task, there are makes of mercury U-tube which can be used in just the same way as this meter and to which all the tables and curves apply equally well. These U-tubes are obtainable from instrument makers.

217. ACCURACY OF METER. The accuracy of the meter must be just as good as, if not better than that of costly recording instruments. They work on the same principle, but the elaborate instruments have pens and counters which impose a load on the indication which limits sensitiveness and impairs accuracy. If the instructions that follow are carefully carried out, the readings given by the home-made meter can be relied on.

218. EFFECT OF PIPE CONSTRICTION ON STEAM FLOW. Steam flow meters are based on the relation which exists between the quantity of steam flowing through a constriction in a pipe and the pressure drop caused by the constriction. The constriction can either be a Venturi tube (or stream-lined throat) or it can be a simple orifice in a sheet metal disc. An orifice causes

the steam to flow in Venturi form. (Venturi was an eighteenth century Italian physicist.) This is shown in Fig. 78 which indicates, by means of diagrammatic mercury columns, the way the pressure changes on either side of the orifice. The principle of the conservation of energy applies to the constricted fluid. (This is known as Bernoulli's theorem and was enunciated by D. Bernoulli, the youngest of many brilliant Swiss brothers, in 1738.) If the stream is constrained to flow faster, it borrows its extra kinetic energy, or velocity head, from the pressure head and so reduces the pressure. The point of maximum constriction of flow—the "vena contracta"—occurs half a pipe diameter downstream of the actual orifice. An interesting point is that just upstream of the orifice the pressure is higher than the average pressure in the pipe. This is because at this point there is a sudden check and the velocity head is converted into pressure head. Below the vena contracta the stream expands and therefore slows down. In so doing it returns much of its velocity head as increased pressure head.

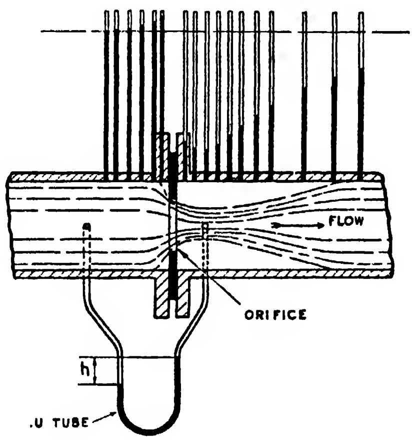


FIG. 78. "VENA CONTRACTA" EFFECT OF FLOW THROUGH AN ORIFICE

If the pressure well upstream of the orifice (one pipe diameter is sufficient to be clear of any disturbance caused by the orifice) is compared with the pressure where the velocity is highest below the orifice (half a pipe diameter, at the vena contracta) we can calibrate this pressure difference against steam flow. The theory and mathematics will not be given here, only sufficient instructions to enable the meter to be calibrated.

219. THE MERCURY U-TUBE. The pressure drop across the orifice is measured by means of a mercury U-tube. If an ordinary glass U-tube is used it will be necessary either to measure the difference in height between the mercury in the two arms, or to measure the height in one arm only. Measuring the difference means moving the scale for every reading. Measuring only one arm halves the height measured and halves the sensitiveness and accuracy.

If, however, the diameter of one arm of the U-tube is made very large compared with the other there will be virtually no movement in the large arm; all the mercury movement will be transferred to the small arm. In the meter

here described a movement of the mercury of 12 in. in the small arm is accompanied by a movement of only $\frac{1}{8}$ in. in the large arm. Although this is apparently an error of 1 per cent., this is not so really, because 1 per cent. of linear movement of the mercury over the upper two thirds of the scale only corresponds to about $\frac{1}{2}$ per cent. in steam flow.

220. POSITION OF ORIFICE IN PIPE. The orifice can be slipped into any convenient flanged joint in the steam pipe. It should be inserted in a long straight length clear of bends, tees, valves, etc. There should be a straight length upstream of the orifice of at least 10, preferably 20, pipe diameters, and at least a straight length of 5, preferably 10, pipe diameters downstream of the orifice. If a globe valve or reducing valve precedes the 10-diameter straight there will be a considerable disturbance of the steam flow in the pipe at the orifice. This disturbance can be largely smoothed out by fitting a grid plate in the flange at the beginning of the straight. The grid plate is just a plate drilled with $\frac{1}{8}$-in. or $\frac{1}{4}$-in. holes. The combined area of the holes should be about half the area of the pipe cross section. There is no need to drill a lot of holes in a plate ; the grid plate can be cut from standard perforated plate. A grid plate should not be used on a low-pressure steam pipe as it produces an appreciable pressure drop.

221. MAKING THE ORIFICE. Instructions for finding the correct size for the orifice are given in Section 230. The orifice is shown in Figs. 79 and 80. It should be cut, if possible, from stainless steel or monel metal, in order that it may retain its sharp edges and exact diameter. The outside diameter should come just inside the bolt circle of the flange so that the flange bolts will centralise it in the pipe. If a lug or handle can be left on the periphery of the orifice plate it makes insertion and removal more convenient. A small hole should be drilled in the bottom of the orifice plate to prevent condensate building up on the upstream side to such an extent as to distort the flow of steam.

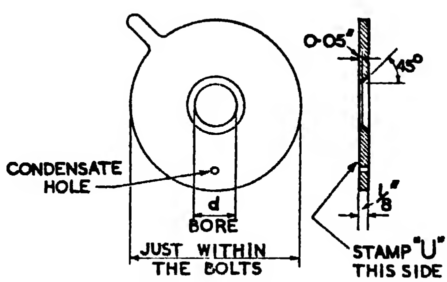


FIG. 79. PIPE METERING ORIFICE PLATE

The orifice plate should be made with care. The accuracy of the actual orifice should be within one thousandth of the orifice diameter. That is to say, a 3-in. orifice should measure 3 in. ± ·003 in. The orifice should be machined with a very sharp tool. Burrs or wire edges must be removed with the greatest care, because the edges should be as clean and sharp as possible. The effect of rounding the edges is to make the meter read low. A burr will make it read

high. Ordinary careless removal of burrs will round the edge quite enough to cause the readings to be 2 or 3 per cent. low. The upstream face of the orifice plate should be stamped with the letter U. Unless this is done it is easy to forget which way it should go into the pipe. The thickness of the orifice plate is given in Table XXXV in Section 230.

Joints on either side of the orifice should be as thin as possible and must not protrude into the pipe—that is to say the inside diameter of the joint rings must be a little greater than the inside diameter of the pipe.

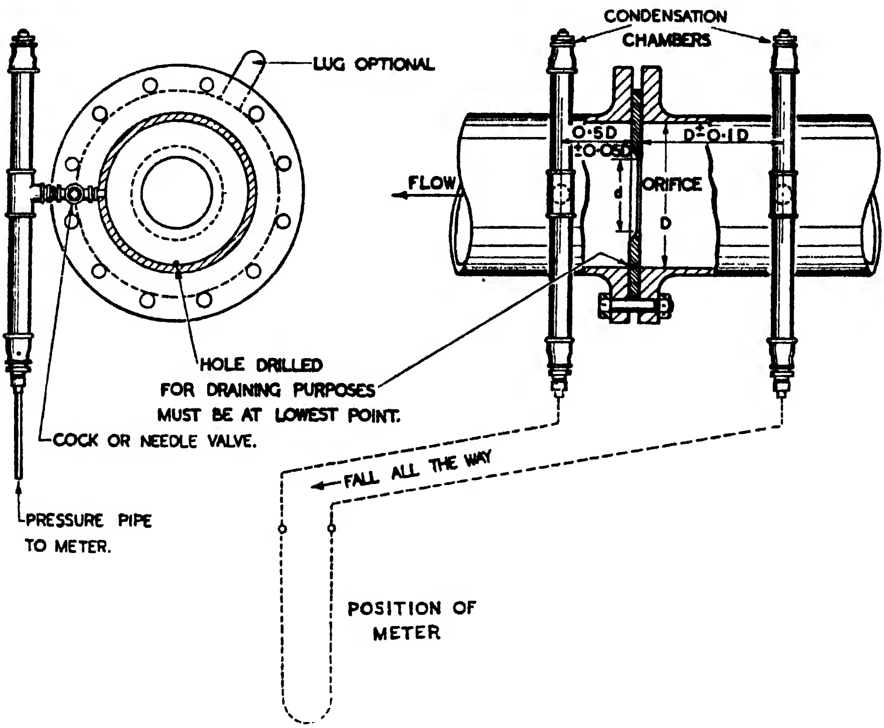


FIG. 80. POSITION OF ORIFICE WITH PRESSURE MEASURING POINTS AND CONDENSATION CHAMBERS

222. PRESSURE CONNECTIONS. Two holes must be drilled and tapped in the steam pipe to suit ½-in. nipples. These holes should never be placed in the bottom of the pipe. The holes should be on the horizontal diameter and should be at the same height. The one hole should be one pipe diameter upstream; the other, half a pipe diameter downstream of the orifice. The position of the holes should be correct to within at least 10 per cent. and allowance must be made for the thickness of the orifice-flange joints when drilling the holes. The arrangements for making an orifice fitting in a vertical main are described in Section 225.

223. CONDENSATION CHAMBERS. The condensation chambers are provided to condense the small quantity of steam needed to make up for the small loss of water that takes place from one or other of the pressure pipes at every change of steam flow. They act as small automatically filled supply tanks which make sure that the pressure pipes to the meter are always completely filled with water.

The components of the condensation chambers are shown in Fig. 81, and their arrangement with regard to a horizontal steam pipe is shown in Fig. 80. They should be fitted vertically with the pressure pipes at the bottom.

If needle valves or cocks are not available for fitting between the condensation chambers and the pipe, small gate valves can be used. If globe valves are used they must be fitted with the spindles horizontal.

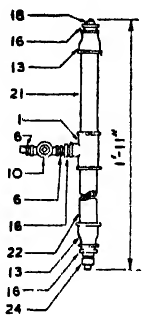


FIG. 81. DETAIL OF CONDENSATION CHAMBERS

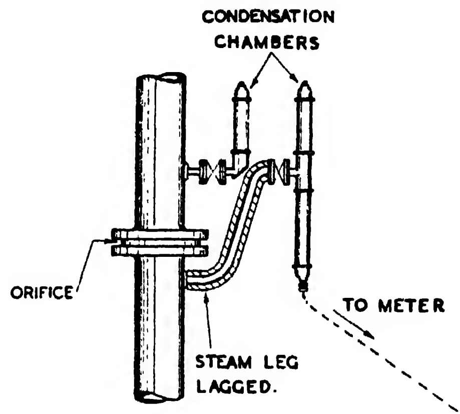


FIG. 82. METERING VERTICAL PIPES

224. ARRANGEMENT OF PRESSURE PIPING. The meter should always be arranged to be below the steam pipe that is being metered, in order to ensure that the pressure pipes are always full of water. If local circumstances demand that the meter be above the steam pipe the pressure pipes must fall for a foot or two from the condensation chambers before rising to the meter. Air release cocks must be fitted in each pipe just above the meter, and much patience and care will be needed to ensure that there is no air in the pressure pipes before a reading is taken. Such an arrangement will almost always be unsatisfactory.

Both pressure pipes should be the same length and should follow the same path. The pipes should fall all the way from condensation chambers to meter. There should be no horizontal runs and, of course, no dips. If they fall continuously any air in the pressure pipes will find its way into the steam pipe and be carried away.

225. METERING VERTICAL STEAM PIPES. If the orifice is to be fitted in a vertical length of steam pipe the arrangement of the condensation chambers and pressure piping must be properly done. Owing to the difference in

level between the upstream and downstream pressure connections the condensation chambers must be placed at the level of the upper connection. A large bore pipe must bring the pressure from the lower connection to the appropriate condensation chamber. This large bore pipe is a "steam leg" and must be

ITEM	MATERIAL	SIZE	No. OFF
1	TEE	$1'' \times 1'' \times \frac{1}{2}''$	2
2	"	$1'' \times \frac{3}{8}'' \times \frac{3}{8}''$	1
3	"	$\frac{3}{8}'' \times \frac{3}{8}'' \times \frac{3}{8}''$	1
4	ELBOW	$\frac{1}{2}'' \times \frac{3}{8}''$	2
5	HEX. NIPPLE	$\frac{3}{8}'' \times \frac{3}{8}''$	2
6	" "	$\frac{1}{4}'' \times \frac{1}{4}''$	4
7	REDUCING NIPPLE	$\frac{1}{4}'' \times \frac{1}{2}''$	1
8	PARALLEL "	$\frac{3}{8}'' \times \frac{3}{8}''$	1
9	COCK	$\frac{3}{8}'' \times \frac{3}{8}''$	1
10	"	$\frac{1}{4}'' \times \frac{1}{4}''$	2
11	SOCKET	$3'' \times 3''$	1
12	REDUCING SOCKET	$\frac{1}{2}'' \times \frac{3}{8}''$	1
13	" "	$1'' \times \frac{1}{2}''$	4
14	" BUSH	$3'' \times 1''$	1
15	" "	$3'' \times \frac{3}{4}''$	1
16	" "	$\frac{1}{2}'' \times \frac{1}{4}''$	6
17	" "	$\frac{1}{2}'' \times \frac{1}{8}''$	2
18	PLUG	$\frac{1}{4}''$	2
19	STAND PIPE 16" LONG	$1''$	1
20	BASE PIPE $3\frac{1}{2}''$ LONG	$\frac{1}{2}''$	1
21	CONDENSATION CHAMBER	$7\frac{1}{2}'' \times 1''$	2
22	" "	$9\frac{1}{2}'' \times 1''$	2
23	UNIONS MALE COPPER	$\frac{3}{4}'' \times \frac{3}{8}''$ O/D	2
24	" " "	$\frac{1}{4}'' \times \frac{3}{8}''$ O/D	2
25	GAUGE GLASS 22" LONG × $\frac{1}{4}''$ BORE		1
26	MERCURY WITHOUT REDUCER		$2\frac{1}{2}$ LBS.
27	SCALE WITH GLASS & $\frac{1}{2}'' \times \frac{1}{16}''$ CLIPS		1
28	BRACKET WITH $\frac{3}{4}'' \times \frac{1}{16}''$ CLIPS		1
29	PACKING WASHER $\frac{3}{16}''$ FILED TO SUIT		1
30	SCREWED WASHER CUT FROM $\frac{1}{2}''$ PIPE $\frac{1}{8}''$ LONG		1
31	SOFT GRAPHITED PACKING		2
32	ADJUSTABLE SCALES		2

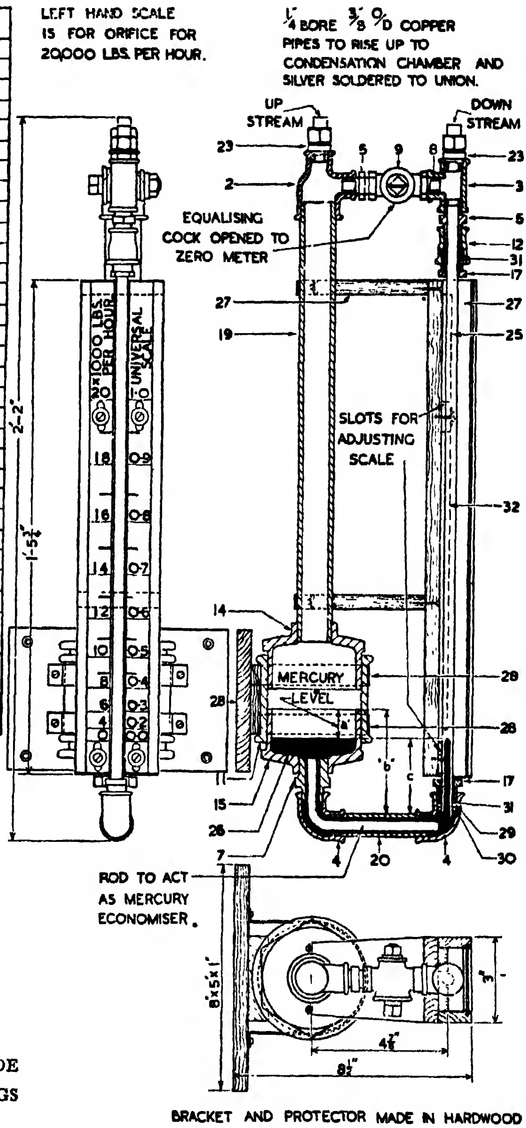


FIG. 83. STEAM METER HOME MADE FROM STANDARD PIPE FITTINGS

sufficiently large to preclude water locking and should be well lagged to prevent condensation. The arrangement is shown in Fig. 82 and applies *mutatis mutandis* to upward or downward steam flow. The pressure connections must of course be taken at the correct half diameter and one diameter distances from the orifice.

The steam leg should be ¾-in. or 1-in. bore. If the steam pipe is of steel there will be too few threads for such a large nipple. A boss should then be welded in the pipe for the lower nipple.

226. CONSTRUCTION OF METER. The construction of the meter is quite straightforward and all the details are shown in Fig. 83. The bent rod labelled mercury economiser simply fills up part of the space that would otherwise be occupied by mercury. As the mercury represents about one-third of the cost of the meter it is worth while economising it as much as possible. Quite an economy can be made by boring out the reducing bush 15, which is the main mercury reservoir, to remove the rounded corner and sloping bottom. This enables the mercury level to be reduced by about ⅛ in. with a saving oı about 15*s*.

The screwed washer 30 is simply to form an abutment for washer 29 which supports the packing. It can be screwed home with a file with a broken tang.

When the metal and wooden parts of the meter have been assembled, the gauge glass is fitted as follows : the downstream union 23 is removed. The glass is then put in through the union hole. As soon as it projects a few inches through reducing socket 12 the packing 31 and the bush 17 can be threaded over the glass tube and assembled finger tight in socket 12. Push the glass tube down to within about 2 in. of elbow 4. Thread the bush 17, the packing 31 and the packing washer 29 on to the glass. Push the glass tube down until it just rests on the inside of the elbow, then pull it up ¼ in. Push in the washer and the packing and screw up the bush 17 finger tight. When the meter is first put under pressure the bushes 17 can be tightened until leakage just stops. Before finally assembling the meter measure the mercury as described in Section 227 below.

227. CONNECTING UP AND FITTING THE METER. Before being used the meter must be tested hydraulically to a pressure 50 per cent. above working pressure.

The pressure pipe from the downstream side of the orifice must be connected to the downstream or gauge glass side of the meter. Likewise the upstream pressure connection must go to the upstream or reservoir side of the meter.

The threads of all connections and unions should be smeared with graphite paste to ensure that the meter can be easily disconnected or dismantled and to ensure absolute tightness.

Set the scale on the meter so that zero on the scale is at the mercury level. See that the slots for adjusting the position of the scale are long enough to enable the scale to be moved about ⅛ in. up or down. Disconnect the pressure pipe from the upstream side of the meter. Make a paper funnel and insert it into the top of the upstream side of the meter. Pour in the mercury. While filling the meter with mercury it should stand in a porcelain or vulcanite dish to catch any mercury that may be spilt. (A large photographic developing dish is convenient.)

The amount of mercury required is found thus. Before the meter is completely assembled fit parts 15, 7, 4, 20 and 4 together. Hold the assembly in the vice making sure that it is perfectly level and plumb in both directions. Pour

mercury into 15 until it is just level with the top of the downstream elbow 4. Put your thumb very firmly over the open elbow 4. Pour in mercury until the rounded corner of 15 is covered. Measure the level of the mercury inside 15. Add sufficient mercury to raise the level by between $\frac{3}{16}$ in. and $\frac{1}{4}$ in. Measure the level of the mercury again. Call this level a. Empty out the mercury and weigh or measure it for future reference. Measure the height of the top of 15 above pipe 20, measurement b. Then $c = b - a$, and is the approximate level for zero on the scale.

When the meter has been assembled, the gauge glass fitted and the mercury poured in, the meter must be filled with water. With both pressure pipes disconnected and the equalising cock open, pour water into the upstream or reservoir arm so that it fills the whole of this arm and runs through the equalising cock into the gauge glass. Fill right up until both arms are full, dislodging any air bubbles on the glass with a wire. Adjust the scale zero to the mercury level.

Connect up the pressure tubing between the meter and the condensation chambers. Take out the plugs 18 from the top of the condensation chambers. Pour water into the condensation chambers until the whole system is quite full of water. The water must only be dripped in to prevent air locking in the pressure pipes. When the whole system is full of water the mercury should still read zero. If it does not, air is locked somewhere in one of the pipes ; the pipe which is air locked being shown by a higher mercury level. Replace the plugs in the condensation chambers.

228. BRINGING THE METER INTO OPERATION. This requires care, but is quite simple. If a mistake is made or the operation is done hurriedly or ham-handedly two things can happen. Much of the water can be blown out of the system ; at worst, some of the mercury can be blown into the steam pipe. Do not put any pressure on the meter unless the glass protector is in position. Put a dish under the meter to catch the mercury if the glass breaks.

Make SURE that the equalising cock is OPEN. Gently open the cock between the downstream condensation chamber and the steam pipe. The whole meter is now under equal pressure. Gently close the equalising cock. If the gauge glass breaks now, the mercury will not be lost. It will just fall into the dish. If the meter is brought into use by means of upstream pressure and the gauge glass breaks after the equalising cock has been shut, the whole of the mercury will be lost. When the meter is under pressure it can be inspected for leaks and the gauge glass glands tightened if necessary (first closing the downstream pressure cock for safety's sake).

As soon as assurance has been made that the meter is free from leaks the cock or valve between the upstream condensation chamber and the steam pipe should be opened VERY GENTLY. The mercury will now rise to a position corresponding to the steam flow.

If both pressure cocks are opened together with the equalising cock shut, some of the water will be pushed into the steam pipe from the downstream side. If the meter is put into operation with the equalising cock shut there is a risk that the sudden surge of pressure will whisk the mercury right out of the meter. This risk is a very definite one when the meter is placed only a foot or two below the steam pipe.

229. THE METER SCALE. The meter and orifice are so proportioned that the mercury will rise 12 in. in the glass for the full flow for which the orifice is suited. The mercury rise—that is, the pressure drop across the orifice—is not a straight line relation with the flow. There is no need to enter into the mathematics here ; they can be found in the text books. The scale can be calibrated from Table XXXIII and Fig. 84. The 1·0 point, at 12 in., on the universal scale is the point of maximum flow for which the orifice has been calculated.

The method of making any suitable scale is shown in Fig. 84.

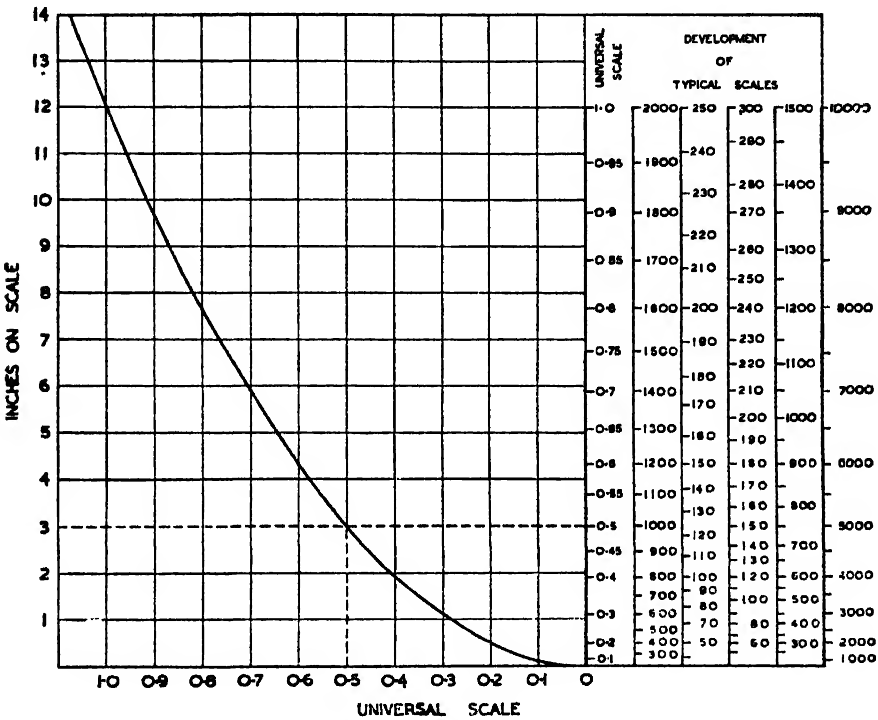


FIG. 84. SCALE CALIBRATION FOR MERCURY U-TUBE STEAM METER

TABLE XXXIII. SCALE FOR MERCURY U-TUBE STEAM METER

UNIVERSAL SCALE	INCHES	UNIVERSAL SCALE	INCHES	UNIVERSAL SCALE	INCHES	UNIVERSAL SCALE	INCHES
·1	·13	·35	1·47	·6	4·32	·85	8·68
·15	·28	·4	1·92	·65	5·04	·9	9·73
·2	·48	·45	2·43	·7	5·88	·95	10·84
·25	·75	·5	3·00	·75	6·76	1·0	12·00
·3	1·08	·55	3·63	·8	7·68	1·05	13·25

230. FINDING THE SIZE OF ORIFICE. From Table XXXIV find the value of the factor K. If the value is not readily obtainable from the Table it can be calculated quite easily or found with the help of a table of square roots. K is the square root of the steam volume in cu. ft./lb. as shown in the steam table.

TABLE XXXIV. FACTOR K FOR ORIFICES

$$K = \sqrt{\text{cu. ft./lb.}}$$

PRESSURE PSI.G.	TEMPERATURE °F.											
	SAT.	300	320	340	360	380	400	420	440	460	480	500
20	3·45	3·56	3·61	3·66	3·71	3·76	3·80					
25	3·24	3·32	3·37	3·42	3·46	3·51	3·55	3·60				
30	3·07	3·13	3·17	3·22	3·26	3·31	3·35	3·39				
35	2·92	2·96	3·01	3·05	3·09	3·13	3·17	3·21	3·25			
40	2·79	2·82	2·86	2·90	2·94	2·98	3·02	3·06	3·10			
45	2·68	2·69	2·74	2·78	2·82	2·85	2·89	2·93	2·96	3·00		
50	2·58	2·58	2·62	2·66	2·70	2·74	2·77	2·81	2·84	2·88		
55	2·49		2·53	2·56	2·60	2·64	2·67	2·71	2·74	2·77	2·80	
60	2·41		2·44	2·47	2·51	2·54	2·58	2·61	2·64	2·68	2·71	2·74
70	2·27		2·28	2·32	2·35	2·39	2·42	2·45	2·48	2·51	2·54	2·57
80	2·16			2·18	2·22	2·25	2·28	2·31	2·34	2·37	2·40	2·43
90	2·06			2·07	2·10	2·14	2·17	2·20	2·22	2·25	2·28	2·31
100	1·97			1·97	2·00	2·04	2·07	2·10	2·12	2.15	2·17	2·20

PSI.G.	SAT.	400	420	440	460	480	500	520	540	560	580	600
120	1·82	1·90	1·93	1·95	1·98	2·00	2·03					
140	1·71	1·76	1·79	1·82	1·84	1·86	1·89	1·91				
160	1·61	1·65	1·68	1·70	1·73	1·75	1·77	1·79	1·81			
180	1·53	1·56	1·58	1·61	1·63	1·65	1·67	1·69	1·71	1·73		
200	1·46	1·48	1·50	1·52	1·55	1·57	1·59	1·61	1·63	1·65	1·67	
225	1·39	1·39	1·41	1·44	1·46	1·48	1·50	1·52	1·54	1·56	1·57	1·59
250	1·32		1·34	1·36	1·38	1·40	1·42	1·44	1·46	1·48	1·49	1·51
275	1·26		1·27	1·29	1·31	1·33	1·35	1·37	1·39	1·41	1·42	1·44
300	1·21			1·23	1·26	1·28	1·29	1·31	1·33	1·35	1·36	1·38
350	1·13			1·13	1·15	1·17	1·19	1·21	1·23	1·24	1·26	1·28
400	1·06				1·07	1·09	1·11	1·13	1·15	1·16	1·18	1·19
450	1·00				1·00	1·02	1·04	1·06	1·07	1·09	1·11	1·12
500	·95					·96	·98	1·00	1·02	1·03	1·04	1·06

Although this Table gives the factor K for pressures up to 500 psi, it must be clearly understood that the pipe fittings meter here described is only suitable for pressures up to 200 psi. For pressures higher than 200 psi a more robust meter must be constructed.

Multiply the value of K found in Table XXXIV by the maximum quantity of steam it is expected will have to be measured. Call this maximum flow

Q lbs./hour. From Table XXXV see whether the steam pipe is large enough for metering by orifice. Table XXXV also shows the recommended thicknesses of orifice plates.

TABLE XXXV. PIPE SIZES FOR METERING STEAM BY ORIFICE

Maximum value of K × Q	6,000	9,000	13,000	24,000	36,000	54,000	95,000	150,000	200,000
Minimum pipe size—inches	2	2½	3	4	5	6	8	10	12
Thickness of orifice plate		1/16"			1/8"			1/4"	

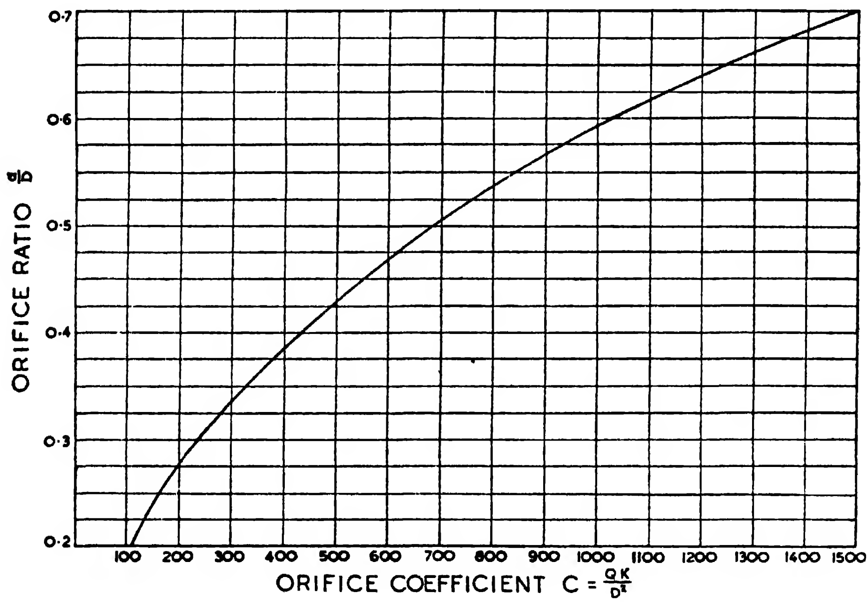


FIG. 85. ORIFICE COEFFICIENT AND ORIFICE RATIO

If the pipe is a suitable size find the orifice coefficient, C, by dividing K × Q by the pipe diameter in inches squared.

$$C = \frac{K \times Q}{D^2}$$

Using this value of C in Fig. 85, find, from the point where it cuts the curve, the value of the orifice ratio $\frac{d}{D}$, where d is the orifice diameter in inches. Multiply

the orifice ratio by D, the pipe diameter, to find the orifice diameter d. Then a flow of Q lbs. steam per hour along a pipe D inches in diameter through an orifice d in diameter will register 12 in. on the mercury gauge of the meter.

EXAMPLE

Estimated maximum flow	Q = 20,000 lb./hour.
Pressure in pipe	33 psi.g.
Temperature	Saturated.
Value of K (from Table XXXIV)	3·0. (This interpolation is doubtful, so it is better to find K by taking the square root of the volume.)

Volume of 33 psi.g. sat. steam is 8·90.

$K = \sqrt{8\cdot 90} = 2\cdot 98.$

$K \times Q = 2\cdot 98 \times 20{,}000 = 59{,}600.$

Minimum pipe diameter (from Table XXXV), 7 in.

Actual pipe diameter, 9 in.

Orifice coefficient $\dfrac{K \times Q}{D^2} = \dfrac{59{,}600}{81} = 736.$

Orifice ratio $\dfrac{d}{D}$ (from Fig. 85) ·513.

Orifice diameter $= \cdot 513 \times 9$ in. $= 4\cdot 617 \pm \cdot 005$ in.

With this orifice diameter in a 9-in. pipe at 33 psi.g. with saturated steam, a flow of 20,000 lb./hour will be shown by 12 in. height on the mercury column.

The minimum straight length of pipe necessary would be 9 in. × 10 = 7 ft. 6 in. upstream of the orifice and 3 ft. 9 in. downstream. If there is space available it is better that these straight lengths be doubled.

231. CORRECTIONS. If the steam pressure or temperature varies appreciably from the value used to determine the orifice, the meter readings will be wrong, but can be readily corrected.

Multiply the meter readings by the value of K which was used to find the orifice size and divide by the value of K actually representing the steam conditions at the time of metering.

For example, if it were desired to use the same orifice as in the example in Section 230 in another 9-in. pipe carrying steam exhausted from a turbine at 40 psi.g. with a temperature of 340° F., we can find the value of K in Table XXXIV as being 2·9. The meter readings must be multiplied by 2·98 and divided by 2·9.

A steam meter registers 1 % low for every 2 % of moisture in the steam. A steam meter registers 1 % high for every 14° F. of superheat in the steam. (See " Efficient Use of Fuel " pp. 220-225.)

232. LIMITATIONS OF METER. No flow meter will work satisfactorily on a pipe with a pulsating flow. For this reason it is very difficult to meter the steam to a slow speed steam engine. If an attempt is made to do so, the meter should be fitted, back along the pipe, as far away from the engine as possible. Even so unless the pipe is very long the meter will not indicate satisfactorily.

The steam after passing through the orifice and the vena contracta never quite regains its original pressure. Some of the energy in the steam is changed into heat by the friction caused by passage through the orifice. This pressure loss is of little consequence in high pressure piping but it may be important with low pressure piping. The pressure loss at maximum flow with this meter is shown in Table XXXVI.

TABLE XXXVI. UNRECOVERED STEAM PRESSURE DROP ACROSS ORIFICE

ORIFICE RATIO $\frac{d}{D}$	LOSS OF PRESSURE DROP	
	PER CENT LOSS OF PRESSURE DROP ACROSS ORIFICE	PSI (12″ MERCURY METER)
·2	95	5·6
·25	92	5·4
·3	89	5·2
·35	86	5·1
·4	83	4·9
·45	79	4·6
·5	74	4·4
·55	69	4·1
·6	64	3·8
·65	58	3·4
·7	52	3·1

From this it will be seen that the effect is more serious in an oversize pipe than in a well-loaded pipe. This is because the introduction of a small orifice in a large pipe creates a much greater relative disturbance than does a slightly smaller orifice in a much smaller pipe.

In the example given in Section 230 the orifice ratio was ·513 which will give a pressure loss of 4·3 psi. Had the pipe been 7 in. in diameter instead of 9 in. the orifice coefficient would have been 1,216, giving an orifice ratio of ·632 resulting in a pressure loss of only 3·5 psi.

This loss of pressure may often be too serious to permit the permanent installation of an orifice in a very low-pressure main. The orifice should however be fitted for a short time and readings taken. It can then be decided whether a permanent meter is desirable and if so, whether its value will outweigh the pressure loss, or whether the measurement, with minimum pressure loss, warrants the installation of a Venturi tube. A properly made Venturi can recover almost all the pressure drop and should always be used, if the cost justifies it, in very low pressure pipes. The foregoing tables and curves do not apply to Venturis. More information, and the formulae for use with Venturis, can be found in British Standard Code B.S. 1042 of 1943, obtainable from the British Standards Institution. B.S. 1042 also gives instructions for using the Pitot tube which is, in some cases, the only device whose use circumstances will allow. The Pitot tube causes no constriction and consequently no loss of pressure.

233. OTHER TYPES OF MERCURY METER. The meter just described is not beautiful. It has been dubbed aptly enough a "plumber's half-holiday". Much neater designs can be adopted if welding and machining facilities are available. An alternative design is given on page 211 of "The Efficient Use of Fuel". This uses the minimum of mercury, but as the level of the mercury goes down in one limb and up in the other it is necessary either to have a moveable scale or to deduct one reading from the other. The latter is inconvenient as it precludes the calibration of the scale into actual steam flow. Very neat, nicely made mercury U-tubes can be obtained from some of the instrument makers.

234. STEAM MEASUREMENT BY MEASURING CONDENSATE. Where steam is used inside a heating surface, the amount of condensate produced is a measure of the steam used. Condensate measurement is generally easy to do, but unless care and knowledge are used it can be very inaccurate. To get an accurate measurement the only instruments necessary are a boy and two barrels.

The trap outlet should be disconnected from the condensate return main and fitted with a swivel pipe or hose so that the discharge can be directed first into one and then into the other of two identical buckets or barrels, whose exact fill has been weighed or measured. These measuring vessels should be filled brimful each time and while one is filling the other must be emptied and the tally marked up together with the time. The size of the buckets or barrels should be as small as possible, compatible with accurate filling and tallying. In this way, provided the times are noted, peaks and valleys can be recorded with a fair approach to the record taken by a costly instrument.

At the discharge of the trap the surplus heat in the condensate causes a flash of steam, see Section 44. This flash of course escapes measurement and must be allowed for. Table IV gives the correction to be applied to condensate that is being discharged at atmospheric pressure. If the heating surface is well drained, if the trap is in good order and is of the right type and if the condensate pipes are well arranged, the steam will give up its latent heat only, and the condensate will be discharged with all its sensible heat. These are the conditions under which Table IV gives the right correction.

The foregoing paragraph assumes that each vessel is fitted with a trap that is working perfectly, that the correct type of trap is fitted, and that the heating surface drains freely into the trap. Unfortunately only too many plants rely on a cracked-open valve to rid their heating surfaces of water, and even the best trap is not infallible especially if it has been applied in a position for which it is not suitable. Faulty traps, faulty condensate draining, cracked-open valves, can cause two faults ; condensate can be held back in the heating surface so that it waterlogs part of the heating surface and slows down the heating while it parts with some of its sensible heat ; or steam can pass in addition to condensate. The empty barrel and Table IV will give wrong results in both these cases.

By using larger barrels partly filled with cold water and by blowing the condensate into the bottom of the barrels, the barrels will contain not only all the condensate, including any flash that might otherwise have escaped, but they will contain the condensate resulting from any steam that has blown direct through. The temperature of the water compared to the amount of

condensate will give us the real picture. This kind of measurement is a combined steam metering and calorimetry and must be done with care, the barrels being weighed, not dipped.

Even when the corrections have been accurately applied, either for flash in the dry bucket or for loss of sensible heat or blowing steam in the cold-water-in-barrel method, both will give wrong results unless the input was dry saturated steam. To get any accuracy at all by weighing or measuring condensate the quality of the input steam must be measured as explained in Section 213.

Two examples of the cold-water-in-barrel method will be given. The first with wet steam, the second with superheated steam. The effect of waterlogging and blowing traps will be seen.

Steam Heated Driers

		Drier 1		*Drier* 2	
	Steam pressure	[by pressure gauge]	27 psi.g.	[by pressure gauge]	27 psi.g.
	Steam temperature ..	[by thermometer]	270° F.	[by thermometer]	292° F.
A	Steam quality	[by calorimeter]	89 % dry	[by thermometer]	22° F. superheat
B	Time of test		32 minutes		19 minutes
C	Sensible heat	[steam table]	239 Btu/lb.	[steam table]	239 Btu/lb.
D	Superheat		—	[·5A]	11 Btu/lb.
E	Latent heat	[steam table]	933 Btu/lb.	[steam table]	933 Btu/lb.
F	Water in barrel at start		137 lb.		121 lb.
G	Temperature of water at start		63° F.		66° F.
H	Heat in water at start ..	[F (G − 32)]	4,247 Btu	[F (G − 32)]	4,114 Btu
I	Water in barrel at end ..		271 lb.		226 lb.
J	Temperature of water at end		111° F.		187° F.
K	Heat in water at end ..	[I (J − 32)]	21,409 Btu	[I(J = 32)]	35,030 Btu
L	Heat increase	[K − H]	17,162 Btu	[K − H]	30,916 Btu
M	Condensate	[I − F]	134 lb.	[I − F]	105 lb.
N	Condensate per hour ..	$[M \frac{60}{B}]$	251 lb./hr.	$[M \frac{60}{B}]$	332 lb./hr.
P	Heat per lb. condensate	$[\frac{L}{M}]$	128 Btu/lb.	$[\frac{L}{M}]$	294 Btu/lb.
Q	Sensible heat given up by condensate	[C − P]	111 Btu/lb.		—
R	Surplus heat in condensate		—	[P − C]	55 Btu/lb.
S	Steam leak from trap* ..		—		5·9%
T	Steam condensed in plant	$[M \frac{A}{100}]$	119·3 lb.	$[M \frac{100 - S}{100}]$	98·8 lb.
U	Useful steam consumption	$[T \frac{60}{B}]$	224 lb./hr.	$[T \frac{60}{B}]$	312 lb./hr.
V	Total steam consumption	[= U]	224 lb./hr.	$[U(\frac{100 + S}{100})]$	330 lb./hr.
W	Wasted steam		—	[V − U]	18 lb./hr.
X	Heat given up in plant	[UE + NQ]	236,850 Btu/hr.	[U (E + D) + DW]	294,725 Btu/hr.

* 1-lb. of condensate "containing" 294 Btu is made up partly of true condensate containing 239 Btu/lb. and partly of live saturated steam (all the superheat will have been taken out) containing 239 + 933 = 1,172 Btu lb. We can find the amount of steam thus :—

Let x = amount of steam.

Then : $1{,}172x + (1 - x)\,239 = 294$.

$$x = \cdot 059 \text{ or } 5\cdot 9 \text{ per cent.}$$

(87664)

The correction for the heat capacity of the barrel has been omitted for simplicity. This correction should of course be applied—see Section 213.

These two examples confirm Jim Hawkins' experience and show what a lot of interesting information can be extracted from a barrel. In Drier I we see that the condensate draining system is so unsatisfactory that half the sensible heat in the condensate is being given up. This must mean that much of the heating surface is waterlogged, greatly reducing the possible output of the drier. In the second Drier, we see that 5·9 per cent. of steam is being wasted by blowing through a faulty trap or through a leaking bye-pass valve.

235. ESTIMATING INJECTED STEAM. If a meter cannot be applied a fairly accurate estimate can be obtained by simple arithmetic. An example will explain the method more easily than an explanation. The following example taken from the author's factory is complicated and shows the kind of thing that may easily be overlooked, namely, the endothermic or exothermic heat.

Example : Calculation of steam used in a process where for local practical reasons metering was impossible.

Process : Washed raw sugar melting.

Vessel : Melter.

The raw sugar has its grosser impurities washed from the crystals in centrifugal machines. Much of the sugar is dissolved in the washed-off syrup—" raw syrup ". The weight of raw sugar is known. The amount and analysis of the raw syrup is measured. The amount of washed raw sugar is unknown except by difference.

Average (round) figures for 26 weeks :—

Weekly melt	9,000 tons.
Raw syrup solids	16·7 per cent.
Washed sugar—by difference	7,500 tons carrying 1 per cent. moisture.
Washed sugar temperature	86° F.
Melting water temperature	194° F.
Melter liquor temperature	158° F.
Liquor density	67·8 per cent. solids.

Heat needed to heat sugar from 86° F. to 158° F. :—

Specific heat of sugar at 86° F.	·303
Specific heat of sugar at 158° F... ..	·337
Average specific heat	·320

$$7{,}500 \times 2{,}240 \times \cdot 32 \times 72 = 387{,}072{,}000 \text{ Btu per week.}$$

Heat needed to heat 1 per cent. moisture from 86° F. to 158° F. :—

$$75 \times 2{,}240 \times 72 = 12{,}096{,}000 \text{ Btu per week.}$$

Heat of solution of sugar—23 Btu per lb. :—

$$7{,}500 \times 2{,}240 \times 23 = 386{,}400{,}000 \text{ Btu per week.}$$

Total theoretical heat required	785,568,000
Add 5 per cent. for radiation	39,278,000
Total heat required	824,846,000 Btu per week.

Water in liquor :

$$\frac{32 \cdot 2 \times 7{,}500 \times 2{,}240}{67 \cdot 8} = 7{,}978{,}761 \text{ lbs.}$$

As water is at 194° F. and liquor is to be at 158° F. each pound of water can provide 36 Btu.

The melter blowers are supplied with open steam at 20 psi.g. measured by calorimeter to be 6·4 per cent. wet.

Each pound of blower steam therefore contains 228 Btu of sensible heat and 93·6 per cent. of 940 of latent heat or a total heat of 1,108 Btu.

Water to be supplied is :—

(Water in liquor) − (Moisture in sugar) = 7,810,761 lbs.

Heat to be supplied is 824,846,000 Btu. Let the amount of steam be x and the amount of water be $(7{,}810{,}761 - x)$. Then :—

$$1{,}108x + (7{,}810{,}761 - x)\,36 = 824{,}846{,}000$$

STEAM USED : $x =$ 507,144 per week.

Hours melting per week :—136

STEAM USED PER HOUR = 3,729 lbs.

Such an estimate of the steam used for direct injection is only accurate if all the steam is condensing. As has been explained in Section 176 superheated steam may pass through uncondensed. Even if the steam were saturated it might acquire superheat owing to wiredrawing in the nozzle or blower, as has been explained in Section 48. The amount of superheat produced by wiredrawing is given in Table VI. This wiredrawing may even occur with wet steam. The water droplets may be scrubbed out of the steam by the liquid being heated and the steam thus dried may expand sufficiently to wiredraw itself. If any of these things are happening more steam will in fact be used than calculation will indicate. It is far better therefore, somehow or other, to get an orifice into the pipe, even if it is a tangle of bends, and have a shot at measuring the flow.

236. A SIMPLE WATER METER. Condensate at or below atmospheric pressure, or the flow of water into a process tank, or into a boiler feed tank, or out of a condenser, can be measured by means of the crude metering tank described by Webre (Section 805) and shown in Fig. 86. Its construction is self-evident. The orifice is the only part that need be accurate. The orifice should be made of the same shape, and with the same care, as the steam meter

orifice shown in Fig. 79 and described in Section 221. As it is impossible in an ordinary factory to produce an orifice of ¾-in. diameter or less to an accuracy of ± ·001 of the diameter, too much trust must not be placed on readings taken from very small orifices. Their flow is, however, very easily verified by the boy and bucket method.

The amount of flow through the orifice is proportional to the square root of the head *h*. In order that the flow through the orifice be reasonably free from disturbance, the flow into the orifice tank should be spread by means of the perforated spreader shown. The orifice should be chosen to be of such a size that the head *h* does not fall below 1·5 times the diameter of the orifice. The balance pipe is essential to prevent unequal pressure above and below the head and to enable the device to work under the vacuum that may exist in the atmospheric pipe of a jet condenser. It should be at least half the diameter of the orifice. It does not matter how big it is. The outlet from the sump must be large and unrestricted so that the sump cannot fill up.

Table XXXVII shows the flow, in pounds of water per hour, through various sizes of orifice for various heads.

TABLE XXXVII. FLOW OF WATER IN LB. PER HOUR THROUGH AN ORIFICE

HEAD *h* INCHES	ORIFICE DIAMETER *d*.													
	¼"	⅜"	½"	⅝"	¾"	⅞"	1"	1¼"	1½"	1¾"	2"	2½"	3"	4"
3	183	411	730	1,141	1,643	2,236	2,920	4,563	6,570	8,943	11,681			
4	211	474	843	1,317	1,897	2,582	3,372	5,269	7,587	10,327	13,488	21.075		
5	236	530	942	1,473	2,121	2,886	3,770	5,890	8,482	11,545	15,080	23,562	33,929	
6	258	581	1,032	1,613	2,323	3,161	4,129	6,452	9,290	12,645	16,518	25,806	37,161	66,064
7	279	627	1,115	1,743	2,509	3,416	4,461	6,971	10,038	13,662	17,845	27,882	40,150	71,378
8	298	671	1,192	1,863	2,682	3,651	4,768	7,450	10,728	14,602	19,072	29,800	42,912	76,288
9	316	711	1,265	1,976	2,845	3,873	5,058	7,903	11,380	15,490	20,232	31,613	45,522	80,928
10	333	750	1,333	2,082	2,999	4,082	5,331	8,330	11,995	16,327	21,325	33,320	47,980	85,298
11	350	786	1,398	2,185	3,146	4,282	5,592	8,738	12,583	17,127	22,370	34.953	50,332	89,479
12	365	821	1,460	2,281	3,285	4,472	5,840	9,125	13,141	17,886	23,361	36,502	52,563	93,445
13	380	855	1,520	2,375	3,420	4,655	6 080	9,500	13,679	18,619	24,319	37,998	54,717	97,275
14	394	887	1,577	2,464	3,549	4,830	6,309	9,859	14,195	19,321	25,236	39,431	56,781	100,944
15	408	918	1,632	2,551	3,673	4,999	6.530	10,203	14,692	20.000	26,120	40,812	58,769	104,478
16	422	948	1,686	2,634	3,794	5,163	6,744	10,538	15,174	20,654	26,976	42,150	60,696	107,904
17	435	978	1,738	2,715	3,910	5,322	6,951	10,862	15,641	21,289	27,806	43,446	62,562	111,222
18	447	1,006	1,788	2,794	4,024	5,477	7,154	11,178	16,096	21,908	28,615	44,711	64,383	114,459
19	459	1,033	1,837	2,871	4,134	5,627	7,349	11,483	16,536	22,507	29,397	45,933	66,143	117,588
20	471	1,060	1,885	2,945	4,241	5,773	7,540	11,781	16,965	23,091	30,159	47,124	67,858	120,637
21	483	1,087	1,932	3,018	4,346	5,916	7,727	12,073	17,386	23,664	30,908	48,293	69,542	123,631
22	494	1,112	1,977	3,089	4.448	6,054	7,907	12,355	17,792	24,216	31,629	49,421	71,166	126,517
23	505	1,137	2,022	3,159	4,548	6,191	8,086	12,634	18,194	24,764	32,344	50,538	72,775	129,377
24	516	1,162	2,065	3,226	4,646	6,324	8,260	12,906	18,584	25,295	33.039	51,623	74,337	132,155
25	527	1,185	2.108	3,293	4,742	6,454	8,430	13,172	18,968	25,817	33,720	52,688	75,870	134,880

Any orifice for this type of meter can be calibrated from the following formula :—

$$Q = 1{,}686 \times d^2 \times \sqrt{h}.$$

Where Q = flow of water in lb. per hour.

d = diameter of orifice in inches.

h = head above orifice in inches.

If the flow is fairly steady, that is to say if the head does not often drop much below about 6 in., this simple device gives reliable readings. It should, however, be checked by the boy and bucket.

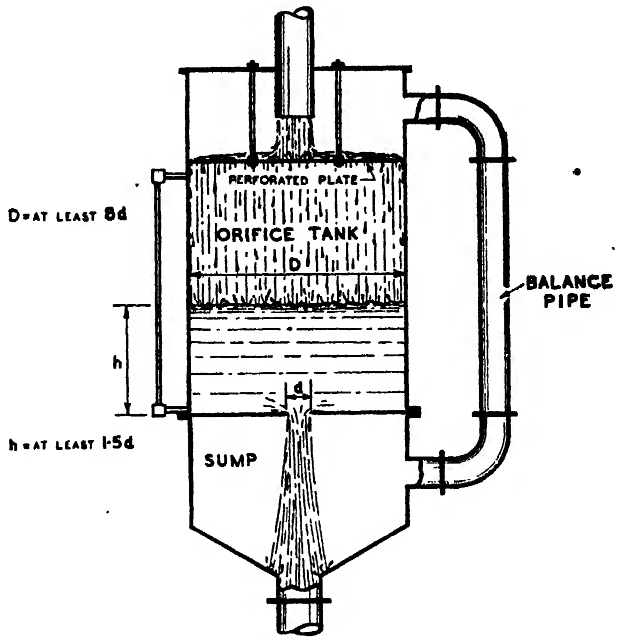


FIG. 86*. SIMPLE ORIFICE WATER METER

237. V-NOTCH METER. The orifice water meter described in Section 236 will not handle widely varying flows, especially flows that might be insufficient to fill the orifice. For widely varying flows a V-notch meter may be more convenient, though even this meter will not be really satisfactory on trickles.

The great advantage of the V-notch meter is that as the flow gets smaller, the measuring orifice, or V-notch, automatically gets smaller too. A V-notch meter can be home-made, and is simple and robust.

Fig. 87 shows a V-notch meter. It consists of a long tank provided with one or more baffles which prevent disturbance of the water surface. The water flows over the V-shaped weir and the flow can be calculated from the height of the liquid above the point of the V.

The actual notch plate should be made from rust proof metal sheet, brass, monel, nickel, or stainless steel, not more than $\frac{1}{32}$ in. thick. The V must be accurately cut and the cut edges must be quite square and sharp. The inside or water face of the notch plate should be quite smooth.

The level of the water can be measured by means of a float or a hook gauge. The float is automatic ; the hook gauge must be hand-set, but is probably more accurate. The float, if fitted, must be well clear of the water flowing into the notch.

* By permission of the Reinhold Publishing Corporation.

The following relations should be used to design a V-notch tank.

The maximum flow level, h, should not exceed $\frac{3}{4}$ of H, the height of the V.

The weir tank, immediately upstream of the notch, should be about 7H wide and 7H long. The baffle, upstream of the weir tank should extend downwards to a level below the point of the V.

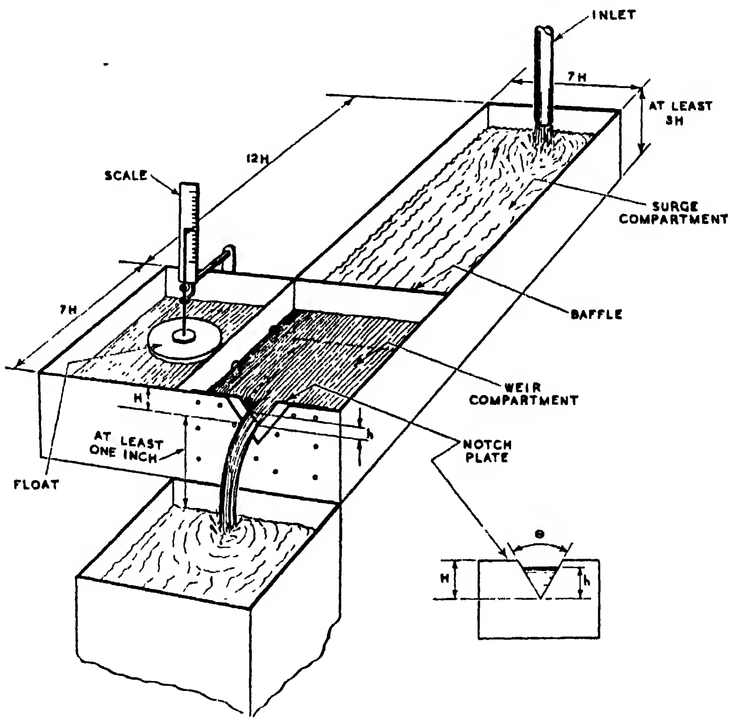


FIG. 87. V-NOTCH WATER METER

The surge compartment into which the water flows should be about 12H long.

The tank should be at least 3H deep.

The flow from the V should be free and should fall 2 or 3 inches below the point of the V. (The absolute minimum fall is 1 in. below the point of the V.)

The float should be situated in a quiet backwater tank, close to the V but sufficiently away from it to prevent the float obstructing the flow into the V.

It is impossible to give tables for reading the flow. The relation changes so rapidly, that the table would be too large for inclusion here.

Table XXXVIII gives the approximate flows in gallons per hour for various notch angles, so that the most suitable angle and tank size can be chosen.

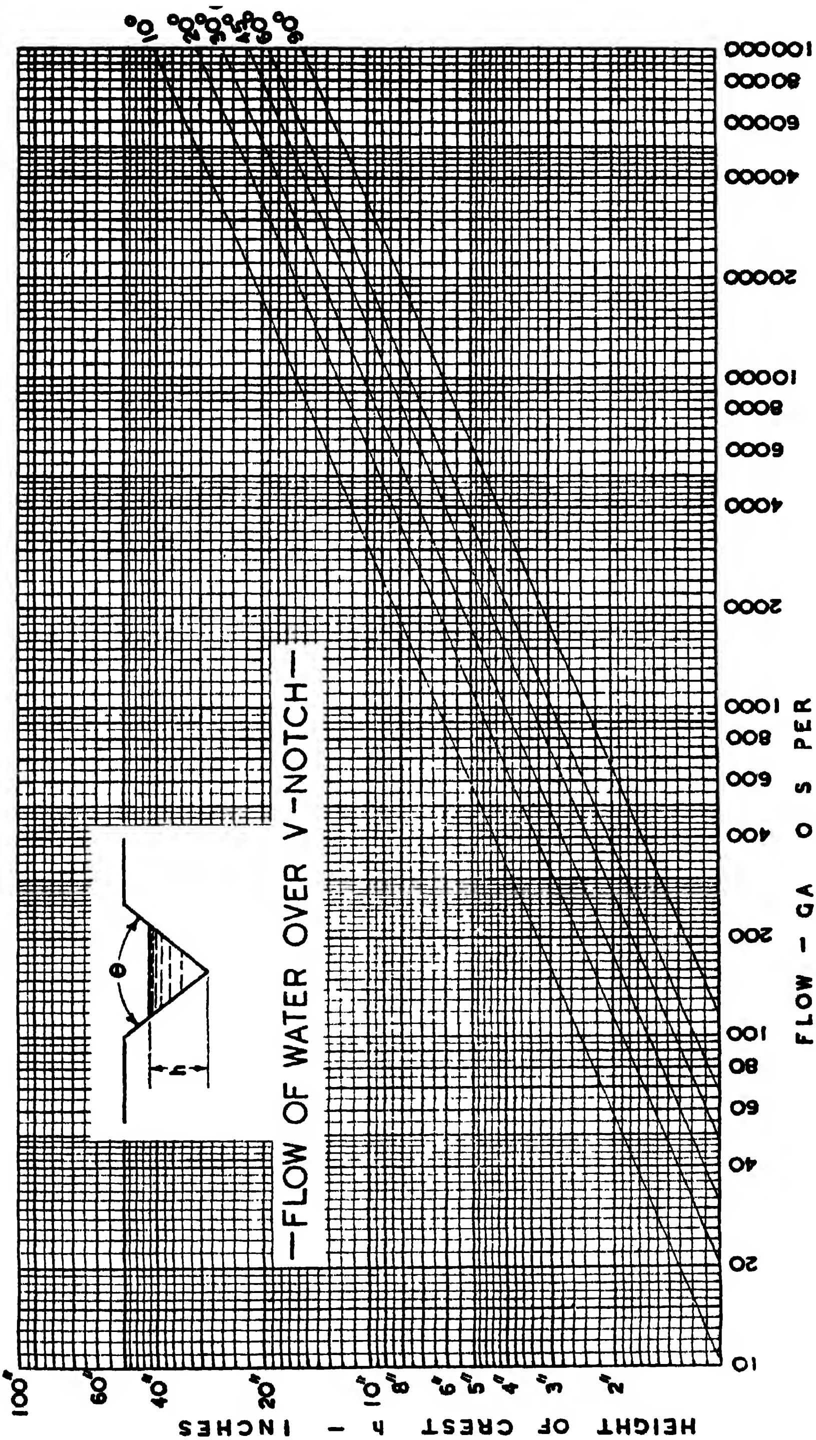


FIG. 88. OF WATER THROUGH NOTCH

TABLE XXXVIII. WATER FLOW THROUGH V-NOTCH METER. IN GALLONS PER HOUR

LIQUID CREST HEIGHT INCHES	NOTCH ANGLE θ					
h	10°	20°	30°	45°	60°	90°
1	10	21	32	49	68	117
2	58	116	176	272	378	654
4	324	649	984	1,517	2,110	3,645
8	1,806	3,622	5,490	8,460	11,770	20,340
16	10,070	20,210	30,620	47,210	65,680	113,500

The formula for flow is :—

$$\text{Gallons per hour} = Kh^{2 \cdot 48}.$$

Where h is the crest height in inches.

The value of K varies with the angle θ of the notch and the values are as follows :—

θ	K
10°	10·40
20°	20·85
30°	31·61
45°	48·73
60°	67·8
90°	117·1

Fig. 88 shows the flow for various angles of V. This chart is probably too small for practical use. A large chart can easily be made by working out two widely spaced results from the formula :—

$$\text{Gallons/hour} = Kh^{2 \cdot 48}$$

plotting them on log log paper when a straight line relates crest height with flow.

Rotary or positive displacement meters give excellent service on clean cold water or clean non-corrosive cold liquids. For hot liquids special meters are required and cold meters are entirely unsuitable. Unfortunately the makers seldom cast COLD or HOT on the bodies of their meters. It should always be assumed that an unmarked meter is not suitable for hot water.

The metering of hot water must be looked on as a distinctly difficult task. The meter must be well chosen and carefully serviced. It is extremely difficult to meter dirty or corrosive liquids, especially if they are hot.

Meters should, wherever possible, be inserted in a length of *straight* pipe in an *accessible well-protected* place and should be provided with a bye-pass to encourage examination and maintenance.

238. MEASUREMENT OF PROCESS MATERIALS. It is of little use carefully measuring steam unless the steam used in a process can be related to the throughput of process material.

Process materials can sometimes be measured by volume but, generally, weighing is the only satisfactory measurement. Volume measurement is, however, so convenient that it is often used. There may possibly be some cases where volume measurement of solids will be accurate, but in the author's experience, limited it is true principally to coal, sugar and charcoal, volume measurement is at best a crude approximation.

Let us take an example. Granulated sugar is at first sight an homogeneous standard article, but a given weight will vary in volume if any of the following qualities change :—

Temperature.
Moisture
Trace impurities
Crystal size.
Size groups.
Presence of adhering sugar dust.
Presence of small crystal agglomerates.
Perfection of crystal shape.
Vibration during filling of the measure.
Height of drop into measure.

The principal factor governing the volume of a given weight of sugar of a particular crystal size seems to be the friction between adjacent crystal faces.

Tapping or vibrating will often greatly reduce the bulk of a granular material but the effect will vary with variation of other characteristics.

Powders can vary tremendously in bulk. After a given weight has been filled into a measure its volume may contract slowly and gradually or it may, after a variable delay, contract suddenly. This contraction may be due to air entangled between the particles, or may be due to electric charges, or to both.

Provided the material to be measured is fairly uniform in size and quality, devices such as the Lea Coal Meter which measure coal volume will work within 2 or 3 per cent. The volume/weight ratio should be checked very frequently, say every few hours.

If the specific gravity and temperature are constant, viscous liquids and homogeneous pastes can be measured by volume, if means are provided for correcting for the adhering residue on the measure. A piston which operates on a constant stroke will give accurate results provided it is quite certain that the cylinder is quite full before each stroke.

Volume measurement, where it can be done, is cheap and simple—much cheaper and simpler than weighing, but the material must be suitable and the conditions favourable or the results will be very untrustworthy.

Whatever the material, there is only one real way of measuring and that is by weighing.

There are continuous weighing machines which fit into conveyor bands. These are only satisfactory for crude weighings and require careful attention, cleaning, and very frequent checking.

The measurement of damp, sticky materials such as raw sugar, suspensions of solids in liquids, pastes which cannot be volume-weighed, etc., present great difficulty.

There seems to be only one satisfactory method, namely to weigh a quantity and then to tare-weigh the material left sticking to the weigh pan after each weighing.

The weighing of liquids having considerable stickiness, or solutions liable to cause incrustation on the weigh pan, can be done with considerable accuracy on self-taring scales. There are several kinds. In one make the scale-pan, after weighing and emptying, is equipoised by a sliding weight. In another type some liquid is retained in the weigh tank so as to give a constant tare weight.

The measurement of air flow and humidity is most important for checking the performance of driers of all kinds.

The measurement of air flow is, in practice, extremely difficult. Accurate measurements can be obtained from a venturi tube or orifice, but it is hardly ever possible to fit a venturi, and an orifice offers too much resistance. The pitot tube is usually the only possible measuring device, but pitot tube readings must be taken at many points over the cross section of the duct. British Standard Code B.S. 1042 : 1943 (obtainable from the British Standards Institution, price 12s. 6d.) gives particulars of pitot tubes and instructions for their use.

Some information on humidity and drying is given in Chapter XXV of "The Efficient Use of Fuel". More information on humidity and on the use of psychrometric charts is given in Chapter XVIII of Faber's book on Heating and Air Conditioning—see Section 805.

239. THE ENGINE INDICATOR. This instrument is of purely specialist application. It records what is going on inside a cylinder in which a piston is reciprocating. It is perhaps unique in the amount and diversity of the information it can give ; consequently it deserves the lion's share of a chapter to itself.

* * *

CHAPTER 7

THE EFFICIENT OPERATION OF ENGINES

This said, he to his engine flew,
Plac'd near at hand in open view.
SAMUEL BUTLER. *Hudibras.* 1663.

IN Chapters 2 and 3 the conditions that govern the efficiency of steam cycles have been considered in detail. The amount by which an engine or turbine falls short of the ideal cycle efficiency has also been discussed. These shortcomings from the ideal were minima ; in practice they are often greatly exceeded. The steam engine and steam turbine are willing horses. They will continue to work in a manner which is satisfactory to outward appearance, when, in fact, they have developed faults which may increase their steam consumption by as much as 50 per cent.

The steam consumption of a turbine is easy to measure. Every turbine should have a steam meter in its pipe. The steam consumption of a reciprocator is often very difficult to measure by meter. Increase in the steam used by an engine may therefore often grow unnoticed. It requires time and effort to determine whether an engine is performing as it should, and the bulk of this Chapter is devoted to ways and means.

240. EFFICIENCIES OF ENGINES AND TURBINES. At best the steam engine or turbine taken by itself is a very inefficient machine. It is only by combining it with a heating process that efficient working is attained. Some comparisons on a coal consumption basis were given in Section 118. These are set out again, with a few additions, on an overall thermal efficiency basis in Table XXXIX. From this sorry table we see that it is of urgent importance that every possible effort be made to scratch together every little bit of efficiency that can be collected.

241. THE PUMPING STEAM CONSUMPTION OF ENGINES. At the beginning of the exhaust stroke the cylinder is full of steam at exhaust pressure. This steam is discharged by the advancing piston until an amount of steam equal to the swept volume has been exhausted. So that there must always be a minimum steam consumption per minute equal to the swept volume of the low pressure cylinder times the number of strokes per minute. The engine acts as a pump which always pumps a minimum amount of steam equal to the cylinder capacity at the exhaust or back pressure. This pumping action accounts almost entirely for the increased steam consumption shown with high back pressures and for the very large variation in steam consumption with variation of load shown by back pressure engines. This point is important and can be better appreciated by taking extremes. If the back pressure were raised almost to the admission pressure the engine would do virtually no work but would simply use its capacity to pump out all the steam at the high back pressure. If the exhaust pressure were an almost perfect vacuum a minute weight of steam would be sufficient to fill the cylinder at the very low exhaust pressure and the weight of pumped steam would be negligible. Table XL shows the amount of this pumped steam for a small simple engine and for a large compound engine.

TABLE XXXIX. APPROXIMATE OVERALL THERMAL EFFICIENCIES OF ENGINES AND TURBINES

(Coal to Useful Power Produced)

TYPE OF MACHINE	EFFICIENCY
	Per cent.
Back pressure turbine, efficient, 5,000 kW	73
Back pressure engine, bad, 100 kW	30
Condensing turbine, 100,000 kW	31
Average British grid (1957)	25
Condensing turbine, medium pressure, 5,000 kW ..	15
Condensing engine, industrial, 1,000 H.P.	9
Colliery winding engine, good	8
Express locomotive	7
Shunting locomotive	4
Colliery winding engine, bad	3
Non-expansive non-condensing pump	2

TABLE XL. ENGINE STEAM CONSUMPTION DUE TO PUMPING AGAINST BACK PRESSURE

	DOUBLE ACTING SINGLE CYLINDER HIGH SPEED SIMPLE	DOUBLE ACTING LOW SPEED COMPOUND
Diameter of low-pressure cylinder	8¼ in.	36 in.
Stroke	10½ in.	48 in.
Swept volume V	·324 cu. ft.	28·276 cu. ft.
Engine speed N	350 r.p.m.	75 r.p.m.
Pumped volume V × 2N	226·8 cu. ft./min.	4,241·4 cu. ft./min.
Steam consumption due to pumping when back pressure is		
20-in. vac.	181 lb./hour	3,390 lb./hour
15-in. „	265 „	4,950 „
10-in. „	347 „	6,500 „
Atmos.	508 „	9,495 „
5 psi.g.	667 „	12,500 „
10 „	825 „	15,425 „
15 „	980 „	18,340 „
20 „	1,133 „	21,295 „
25 „	1,283 „	24,120 „
30 „	1,440 „	27,075 „
40 „	1,740 „	32,835 „
50 „	2,040 „	38,100 „

From Table XL it will be seen that an engine working against a high back pressure will be very extravagant in steam at light loads. This generally is of little consequence except during start-up and shut-down. In most factories the back pressure engine is seldom lightly loaded. It is generally unable to supply the power needed. It is important that engines working against a high back pressure should be of such size that they are working well loaded. It is much better to work one engine right up to full load or even overloaded than to run two or more engines on part load. This was very clearly shown in the Willans Line for two engines in Fig. 39, Section 122.

The turbine has a similar action, though it is less easy to understand. The spaces between the turbine blades are a series of reducing orifices. The blades and nozzles are all the wrong size at very light or no load and the steam is very inefficiently used. In large sizes it may be that the light load steam consumption is less than that of the corresponding reciprocator. As in a reciprocator the higher the back pressure the greater the steam consumption due to the lower pressure drop.

242. THE FRICTIONAL STEAM CONSUMPTION OF ENGINES AND TURBINES. The losses due to friction vary greatly, but are seldom very large. In poor cases the mechanical efficiency at full load may be 90 per cent. while some engines exceed 97 per cent.

In a lubricated bearing most of the resistance to be overcome is due to the viscosity of the oil. It is therefore of great importance that the oil be well chosen. Lubricating oils have suffered from high pressure salesmanship. Cheap oil is generally dear oil.

In a turbine there are many fewer bearings than in a reciprocator, and these bearings are not subjected to heavy accelerating and retarding forces at every revolution. It might be thought that the frictional loss in a turbine would be much less than that of a reciprocator. This is not so, however, particularly at light or no load.

If a turbine with several stages is running light almost the whole of the pressure drop will occur over the nozzles of the first wheel. The remaining blades just churn around in the steam. If the steam is dense, due to high back pressure, the friction produced by this churning may be really appreciable. The lighter the load the greater the number of idle wheels and the greater the churning. This is one of the reasons for the disappointing results often obtained from mixed pressure turbines. When working on the low pressure the high pressure wheels are idle and just churn in steam at the input low pressure. To keep them cool it may be necessary to admit a little high pressure steam. Contrariwise, when working on high pressure the low pressure wheels are underloaded and do considerable churning. It is dangerous to try to give a definite figure for this churning loss. It varies with the conditions and with each particular design of machine, but it is often large enough to do more than counterbalance the reduced bearing friction possessed by a turbine compared to a reciprocator.

243. LOSS BY RADIATION. This is a loss common to turbines and engines. As a turbine for a given power is smaller than the corresponding reciprocator the losses by radiation are smaller for a turbine than for a reciprocator. Another

reason for greater losses from a reciprocator from radiation is that the piston rod goes into the cylinder right into contact with the hot steam and then comes out for a breather every stroke. Generally speaking turbines are better lagged than reciprocating engines, but, compared to a high pressure steam pipe, the lagging of turbines is deplorable. An inch or two of indifferent compound ; the flanges perhaps bare. This iniquity is then covered up by a nice planished steel cover. Engines are even worse. All steam pipe flanges in an economical factory are lagged. How often has anyone ever seen the flanges of an engine cylinder lagged ? How many cylinder heads are lagged ? Why ? It is feared that poor old grandfather is to blame again. He did not lag his cylinder flanges and covers, so we do not. IT IS MORE IMPORTANT TO LAG AN ENGINE CYLINDER THAN ANY OTHER PIECE OF PLANT.

244. GLAND LEAKAGE. From a reciprocating engine, gland leakage is visible and obvious. It makes the engine room hot and uncomfortable, consequently it is seldom allowed to get bad.

The gland in a turbine is usually a labyrinth in two or more main stages. Only the last stage is, as a rule, open to the engine room. In condensing turbines, the intermediate stage of the high pressure gland is often led into the low pressure end of the turbine in good installations ; straight into the condenser in bad plants. In back pressure turbines the makers often insist that the glands should discharge to atmosphere. This is very wasteful, but it has the undoubted advantage that the waste is visible and audible.

Gland leakage in a turbine can be very severe if the gland gets damaged. The turbine will continue to run apparently satisfactorily, and, unless the gland steam can be seen, heard or measured in some way, a very large waste can go on undetected for months. Turbine glands can easily get damaged if the turbine gets distorted by expansion, particularly at start-up. In a back pressure turbine the casing is often racked by the exuberant expansion of a huge exhaust pipe and the resultant distortion may easily cause the delicate labyrinth to rub and be partly destroyed. This usually happens quite inaudibly and discreetly, so a watch should regularly be kept on gland leakage. (In 1944 the damaged glands in the author's turbine were passing about 8,000 lb. steam per hour ; nearly 10 per cent. of the turbine steam.)

Turbine gland leakage is not by any means always complete waste, but it always represents useless increase of entropy. The exact significance of turbine gland leakage must be considered. The high pressure gland is at the pressure of the first-wheel chamber or cell, something considerably less than stop-valve pressure. How does the steam that passes through the high pressure gland get to the gland ? Does it all go through the impulse wheel and round over the top ? Or does it sneak out of the nozzle straight into the gland without first doing a job of work in the wheel ? Probably a bit of both. The steam that passes out of the low pressure gland has all passed through all the turbine wheels and has generated all the power that it can. (We are discussing the low pressure gland in a back pressure turbine.) All this gland steam can be used for heating at atmospheric pressure or a little higher (in the author's factory all the gland leakage goes into the 10 psi main). So the loss in the low pressure gland is the difference in pressure between the back pressure and the pressure at which the

gland steam is used. The loss in the high pressure gland steam is probably about 70 or 80 per cent. of the power producing capacity of the steam. It can only have done work in the first wheel, it is entirely lost to the remaining stages, and some of it may never have passed through the first wheel at all.

Gland steam can be metered by means of an orifice and mercury U tube as described in Sections 215 to 233. In order to reduce the throttling effect of the orifice, the orifice size should be calculated for a very much greater flow than the expected amount of the gland leak. Where gland steam is piped direct to the condenser, a thermally deplorable deed, it is very important that it should be periodically metered as the whole of the heat is completely lost.

245. CYLINDER CONDENSATION. This is by far the greatest loss in most reciprocating engines. It is rather complicated but merits detailed consideration.

Assume that the engine is hot and is being supplied with saturated steam. During the exhaust stroke the cylinder walls, cylinder head, valve passages and piston head are cooling down to exhaust temperature. After the exhaust valve has shut and compression is taking place no further loss of heat to the exhaust can take place. Compression raises the temperature of the compressed steam but most of this heat is given up to the metal of the cylinder. When the admission valve opens the high pressure steam enters and at once gives up much of its heat to heating the steam ports, cylinder head and piston head. If the steam is saturated it can only part with heat by condensation. When the piston moves forward it uncovers comparatively cool cylinder metal. So that during admission condensation continues up to cut-off. After cut-off the pressure in the cylinder falls and, although the steam is still trying to part with heat to the cylinder, this is partly compensated by the production of flash steam from the condensate in the cylinder. However, this balance is upset because the steam is parting with the heat that it is converting into work on the piston. This causes further condensation, as has been explained in Chapter 2. When the exhaust valve opens the sudden reduction in pressure causes a flash of steam from the condensate. During the exhaust stroke the now hot cylinder walls readily give up their heat to evaporating the condensate and are thus rapidly cooled down towards the saturation temperature appropriate to the exhaust pressure. At the end of the exhaust stroke much of the condensate has been re-evaporated at the expense of heat in the cylinder walls, cylinder head, piston and valve passages. The process repeats itself at every stroke.

The greater the amount of expansion in the cylinder the greater will be the temperature difference at the beginning and end of each cycle. So that an attempt to get greater expansion in a single cylinder will be accompanied by much greater condensation losses than if the expansion were shared by two cylinders as is done in a compound engine.

The faster the engine runs the less time will there be for the cylinder walls to absorb heat from the live steam and cause condensation, and the less time will there be for the metal to give up heat to the condensate for evaporation during exhaust.

If the admission and exhaust valves can be entirely separated the inlet ports will not be cooled by the outgoing exhaust.

If the cylinder could be kept sufficiently warm to prevent condensation there would be no loss by the re-evaporation of condensate.

All these things have been tried with varying success. By using a small high speed engine in place of a large slow speed machine the condensation losses are reduced, but frictional losses may be greater. It cannot be affirmed that the high speed engine is always more efficient than the low speed engine. By separating the admission and exhaust valves a definite gain has been achieved, variously estimated at between 10 and 20 per cent., but usually the comparison is difficult as engines with separate valves usually have a quick acting valve gear that is anyhow an improvement on the slide valve or piston valve.

By applying a steam heated jacket to the cylinder a gain of between 5 and 25 per cent. has been found. Heat is supplied by the jacket which greatly minimises condensation in the cylinder although it of course causes condensation in the jacket. But the condensate in the jacket cannot be re-evaporated and lost and this is the cause of the economy. Nowadays jackets are seldom fitted for reasons that will be explained in Section 246. The steam jacket, or "case" as he called it, was invented in 1765 by James Watt, who said: "To make a perfect steam engine it is necessary that the cylinder should always be as hot as the steam which enters it."

246. SUPERHEATED STEAM. By superheating the steam the following benefits appear; initial condensation can be eliminated; condensation due to work producing expansion can be eliminated; if there is no condensate present at exhaust there can be no loss due to re-evaporation during exhaust; dry steam during exhaust draws heat less readily from the metal cylinder than does a film of condensate. If the admission and exhaust valves are separate less superheat will be needed than if they share common ports.

Cutting off the jacket steam will probably improve matters if the engine is being supplied with superheated steam. The jacket is provided solely to limit or reduce condensation in the cylinder. If condensation has been prevented by superheating, the purpose of the jacket is gone and all it can do is to offer a larger heat-losing surface than does the unadorned cylinder.

Superheating was first commonly used about 1860 and the economy resulting (up to 50 per cent. saving in steam) was so great that superheating was carried too far. The cylinders of the engines of those days were lubricated with tallow which decomposed at quite moderate temperatures. The result was serious scoring of the valve faces, cylinder bores and pistons, and superheat was practically abandoned in 1870. Increasing admission pressures, the improvement of compound engines and the production of better condensers and vacuum pumps provided easier methods of improving economy than the overcoming of the lubrication difficulties at high temperature. The limitations of metals and of manufacturing technique soon put a brake on pressure increase and by 1900 superheat had come back to find that improvements in oils were available to let it take its rightful place in thermo-economy.

In 1897 Professor W. Ripper carried out his famous experiments with superheated steam on a small 13 H.P. single-acting engine with two cylinders of 7-in. bore taking steam at 100 psi.g. and running at 180 r.p.m. He obtained the results shown in Table XLI.

TABLE XLI. EFFECT OF SUPERHEAT ON STEAM CONSUMPTION OF SMALL RECIPROCATING ENGINE
(Ripper)

APPROX. SUPERHEAT °F.	STEAM CONSUMPTION LB./H.P./HOUR	RELATIVE EFFICIENCY IMPROVEMENT PER CENT	ADIABATIC HEAT DROP BTU	RELATIVE CYCLE IMPROVEMENT PER CENT
Sat.	39·6	—	140	—
100	33·8	15	151	8
150	29	25	159	14
200	25	36	166	19
250	23·4	44	171	22
300	20·1	49	187	33

It will be seen that the actual efficiency improvement due to superheating is about double the amount of the improvement to be expected from the greater heat drop, except with a superheat of 300° F. The inference to be drawn from Table XLI is that somewhere between 250° and 300° of superheat was sufficient to prevent any condensation at all, and beyond this point additional superheat merely increases the available heat drop.

247. DIFFICULTIES WITH SUPERHEAT. Superheated steam is a dry gas. The valves and pistons of engines must be lubricated if superheated steam is to be used. This means that oil will be present in the exhaust steam. If the engine is exhausting to a condenser the oil will coat the condenser tubes and reduce the heat transfer, and the condensate will consist of an oily emulsion. Such condensate is unsuitable for boiler feed, so that unless means can be found for removing the oil, then the use of superheated steam must be abandoned, or the condensate must be wasted, or more cooling water must be used in the condenser, or a lower vacuum must be tolerated. The not very satisfactory methods for removing oil from steam and condensate have been discussed in Section 184.

If the exhaust steam is to be used for process, oil is probably very undesirable. It will coat the surfaces of calorifiers and pans ; it cannot be permitted if the steam is to come into contact with food or textiles

Where superheat is possible, and there are hundreds of factories where the conditions permit its use, it should not be pushed up without expert advice. Valves and glands that are suitable for saturated steam or for moderately superheated steam may be quite unsuitable to withstand highly superheated steam. The oil must be chosen with care. Any of the reputable oil companies will recommend the correct oil for the particular conditions. Temperatures above about 650° F. are not advisable in reciprocating engines.

High superheat is not an unmixed blessing. It is better to use a moderate superheat with reliability and freedom from worry, forfeiting a little efficiency, than to strive for the last 1 per cent. of efficiency and to have continual anxiety and trouble.

The turbine can accept highly superheated steam without any lubrication difficulties. The limits are simply those imposed by reasonable prudence and the desired exhaust state (Sections 79, 80, 100, 145). It is one of the tiresome contrarinesses that the turbine will accept superheat without difficulty whereas the reciprocator makes all kinds of troubles, yet the turbine benefits far less from superheat than does the reciprocator.

248. BENEFITS OF SUPERHEAT. We can summarise the benefits thus: Superheat reduces or even eliminates the greatest loss, cylinder condensation, in a reciprocating engine; Superheat gives an appreciably increased cycle efficiency; it gives proper control of the quality of the exhaust steam—this is desirable for process purposes in order that the process plant may be fed with dry saturated steam, and it enables the moisture in the last row of blades in a condensing turbine to be kept at about 12 per cent.; the steam pipes can be smaller because steam velocities can be nearly twice as great with superheated steam as with wet steam (Table XXVII, Section 172).

Another benefit of superheat is in the boiler plant. Superheated steam contains more heat than saturated steam, and this extra heat must be put in in the boiler. This would at first sight seem to call for more coal, but often no more coal need be burnt especially if the plant consists of shell boilers. The superheater is extra heating surface in the boiler which takes more heat out of the combustion gases, so that much, if not quite all, of the extra heat needed may be obtained for nothing.

249. INEFFICIENCY IN BACK PRESSURE MACHINES. In Section 112 it was stated that "it does not matter how inefficient an engine is, provided there is a real use for all the exhaust steam". This statement is true, but it must not be allowed to encourage complacency towards inefficiency in a back pressure machine. In some factories steam is used lavishly with the excuse "It's engine exhaust and would be blown to atmosphere if we didn't use it". Now is that true? A roaring safety valve has a fine psychological effect. Everyone realises that there is waste going on; consequently everyone will try to minimise the waste by keeping the engine or turbine up to the mark, by reducing the power load wherever possible, by keeping the back pressure as low as possible—see Section 125.

A back pressure machine should be kept in as efficient a state as possible, thus giving the process every encouragement to economise in the use of steam. If the steam/power ratio is such that the process cannot absorb all the exhaust from the power house, it is even more necessary to maintain the power-producing plant in first-class condition.

If the load is a series of peaks and valleys, it is particularly important that back pressure machines be kept in as good trim as possible so that their steam consumption may be a minimum. In this way the boilers can meet the peaks and the engine exhaust does not blow off in the valleys.

250. SIGNS OF ENGINE AND TURBINE DEFECTS. The metering of steam to turbines presents no difficulty and there is no excuse in any factory for ignorance as to the steam consumption of turbines. Metering of steam to reciprocators, especially of slow speed, is practically impossible, so indirect methods must be used to check any falling off of efficiency.

Any steam plant such as a pumping station, which is working under steady conditions and where the engine is the only steam user, will show defects in the engine as an increased consumption of coal, provided the quality of the coal remains constant and the boiler efficiency does not vary.

Most engines do not work under such conditions, and, unless energetic steps are taken to check engine efficiency, great waste can go on for months undetected. In order to know whether the engine is working efficiently or not it is necessary to know :—

1. How much steam the engine is using.
2. How much steam it ought to use.

Neither of these is at all easy to ascertain.

251. HOW MUCH STEAM SHOULD AN ENGINE USE ? The engine makers should be able to say how much steam the engine should use at various loads. The makers may have gone out of business or the engine may be of foreign make. Advice should then be sought from makers of similar machines. If no information can be obtained from these sources it will be necessary to make your own estimate and some indications of how to go about this are given below. Quite a lot of information has already been given in Section 104. Whether the figure be obtained from the maker or whether it is an amateurish estimate a major difficulty at once arises. Is the engine on full load, three-quarter load, half-load or overload ?

252. WHAT IS THE WORK DONE ? If the engine is driving a dynamo or alternator the switch board instruments will at once show the amount of energy being delivered by the generator. This will be rather less than the power the engine is developing owing to the losses in the generator and in the engine. 1 kW is equivalent to 1·34 brake horse power. There are mechanical and radiation losses in the engine and electrical losses in the generator which vary greatly but which, for purposes of estimate, we can take as being about 20 per cent. for moderate sized machines of 300 kW or so, and say 30 per cent. for small machines of 50 kW and under. By allowing for these losses we can get at the theoretical or indicated horse power. A rough rule is to take 1·5 indicated horse power per kW for a large machine, 1·7 I.H.P. per kW for a fair sized machine and 1·9 I.H.P. per kW for a small machine.

Where the machine is driving a line shaft, a rolling mill, winding coal or spinning textiles the best way is to try to compare conditions with another plant doing the same work by electric motors. At best only a very rough and rather untrustworthy estimate can be obtained.

253. WHAT STEAM IS NEEDED. Having made the best possible estimate of the amount of power being produced by the engine, an estimate can be made of the steam required. This is done by taking the adiabatic heat drop of the

cycle over which the engine is working and correcting for the efficiency ratio that can reasonably be expected. This has been described in Sections 103 and 104, but no indication was there given of the probable efficiencies of very small machines. Fig. 89 gives the adiabatic heat drop in terms of pounds of steam per H.P., to save the trouble of working it out from the steam tables. These curves give the theoretical steam consumption per Internal or Indicated Horse Power in an ideal engine. They must be corrected for the probable losses that will be present in the actual engine. Table XLII gives an indication of the efficiency ratio to be expected from different types of small machines. Tables IX and X gave efficiency ratios for larger machines.

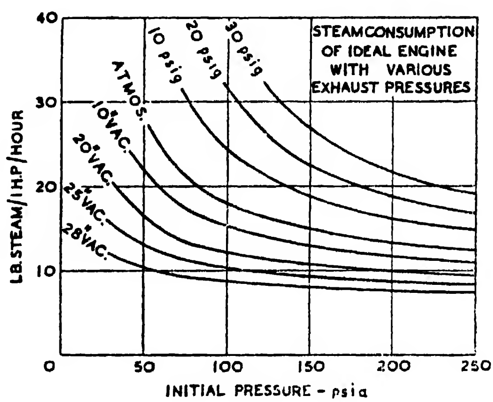


FIG. 89. STEAM CONSUMPTION OF IDEAL ENGINE WITH VARIOUS INITIAL AND EXHAUST PRESSURES

One point of interest in Table XLII is that condensing engines are shown as having lower efficiency ratios than non-condensing engines. This may seem surprising. There must be no confusion between Thermal Efficiency and Efficiency Ratio. The cycle efficiency (and probably the thermal efficiency) of the condensing engine will be higher than that of the non-condensing engine, but the efficiency ratio of the condensing engine will be lower because there is a greater temperature difference between exhaust and admission in the cylinder of the condensing engine ; there will consequently be greater heat losses to the cylinder.

The steam consumption ought to be :—

$$\underset{\text{(From Fig. 89)}}{\text{Ideal consumption per I.H.P.}} \times \frac{100}{\text{Efficiency Ratio}} \times \begin{pmatrix}\text{estimated horse-}\\ \text{power required.}\end{pmatrix}$$

(from Tables IX, X, XLII).

Example I.

A simple engine is estimated to be developing 60 H.P. It is supplied with dry saturated steam at 100 psi.g. and exhausts to atmosphere. What should be the steam consumption in lb. per hour ?

Absolute pressure is 115.

115 cuts the Atmos. curve in Fig. 89 at 17 lb./I.H.P./hr.

Probable efficiency ratio from Table XLII is 46 per cent.

Steam consumption should be : $17 \times \frac{100}{46} \times 60 = 2{,}217$ lb./hour.

Say mechanical efficiency is 85 per cent., then real steam consumption should be $\frac{2{,}217}{\cdot 85} = 2{,}610$ lb./hour.

TABLE XLII. APPROXIMATE EFFICIENCY RATIOS OF SMALL ENGINES

TYPE OF ENGINE	WITHOUT SUPERHEAT	WITH SUPERHEAT
Simple non-condensing up to 50 I.H.P.	43	50
Simple non-condensing up to 300 I.H.P.	60	70
Simple condensing up to 50 I.H.P.	41	48
Simple condensing up to 300 I.H.P.	55	65
Compound condensing up to 50 I.H.P.	55	65
Compound condensing up to 300 I.H.P.	70	75

The figures given in Table XLII, in so far as they can be compared, are rather more optimistic than those in Table X. The figures for the two tables were taken from different authorities.

Example II.

Compound engine driving dynamo. Output 200 kW. Steam supply superheated steam at 185 psi.g. Engine exhausting into condenser at 25-in. vacuum. What should be the steam consumption ?

Absolute pressure is 200.

200 psi.g. cuts the 25-in. curve in Fig. 89 at 8·8 lb./I.H.P./hr.

3 kW is equivalent to 4 H.P. Therefore the engine is developing 267 H.P.

Probable efficiency ratio from Table XLII is 75 per cent.

Probable mechanical and electrical efficiency 80 per cent.

Steam consumption should be : $8{\cdot}8 \times \frac{100}{75} \times \frac{267}{\cdot 8} = 3{,}920$ lb./hour.

254. ESTIMATING THE WEIGHT OF STEAM THAT SHOULD ENTER THE CYLINDER. Another method of finding the steam needed is now described. It does not really find the steam needed, nor does it find the steam actually being used. It is a kind of cross between the two and is given for what it is worth.

If we know at what point of the stroke the steam is cut-off, then the cylinder volume at cut-off should represent the amount of steam that enters the cylinder at each stroke. We must consider the high pressure cylinder only in the case of

compound engines, and all the high-pressure cylinders in the case of multi-cylindered engines. If the pressure of the steam entering the cylinder is known, the weight of this volume of steam can be found from the steam table. The pressure in the cylinder during admission can be found from the indicator diagram (which is described below in Section 259 onwards). The point of cut-off can also be found from the indicator diagram or it can be found as described in Section 273.

The actual weight of steam that enters the cylinder is, however, always greater than this, owing to condensation in the cylinder, unless the steam is highly superheated. If the steam is highly superheated the amount of steam may be less than the volume calls for due to wiredrawing in the valve ports. Allowance, on a rough practical basis, can be made for these variations by multiplying the apparent weight of steam by the following factors :—

Highly superheated steam	·95
Moderately superheated steam	1·0
Slightly superheated steam	1·1
Dry saturated steam	1·25
Wet steam	1·45

Example.

Single cylinder, double acting engine, bore 10 in., stroke 12 in., speed 200 r.p.m. Steam is dry saturated at 85 psi.g. At full load cut-off takes place at half stroke. What is the probable steam consumption ?

Piston displacement up to cut-off : $\dfrac{22 \times 5 \times 5}{7 \times 144 \times 2} = \cdot 273$ cu. ft.

Add 5 per cent. of the volume for clearance $\cdot 273 + \cdot 027 = \cdot 3$ cu. ft.

This then is the apparent volume of the steam entering the cylinder.

The volume of steam at 85 psi.g. is 4·45 cu. ft./lb.

Then ·3 cu. ft. will weigh ·0674 lb.

For dry saturated steam the condensation factor (see above) is 1·25.

Probable weight of steam per stroke is $\cdot 0674 \times 1\cdot 25 = \cdot 0843$ lb.

Steam consumption per hour is $\cdot 0843 \times 200 \times 2 \times 60 = 2{,}023$ lb./hr.

255. FINDING THE ACTUAL STEAM CONSUMPTION. 1. *By Steam Meter.* This is seldom possible except with high speed engines supplied by large steam mains which are sufficiently long to enable the orifice to be inserted far enough away from the engine to be reasonably free from pulsations. Meters can, however, be used with all turbines. It is generally useless to try to measure the consumption of a slow speed engine by meter. It is sometimes suggested that by throttling the pressure pipes leading from orifice to meter the pulsations can be smoothed out. This is true up to a point but the accuracy is sometimes smoothed out as well.

2. *By weighing the condensate.* This method can only be used where the engine is exhausting to a surface condenser or into a process from which the WHOLE of the condensate is returned. It is done by collecting the condensate in buckets or barrels and weighing the amount discharged in a given time.

If weighing is inconvenient measurement can be substituted. If small measuring vessels are used it is possible, by logging time as well as quantity, to get some idea of variations in consumption. Before measuring condensate the condenser should be tested for leaks, as any leakage of condensing water will add itself to the condensate and make the engine steam consumption appear greater than it really is. (See Section 234.)

3. *By measuring the cooling water.* If the engine is exhausting into a jet condenser the condensing steam mixes with the cooling water and it is not possible to measure the condensate as in 2 above. If, however, it is possible to find the rate of flow of the cooling water into the condenser the amount of steam can be estimated approximately from the rate of flow and the rise of temperature.

Example.

The rate of flow to a jet condenser is found by V notch to be 4,000 gallons per hour. The water enters at 60° F. and leaves at 90° F. What is the probable steam consumption of the engine ? Vacuum 26 in.

Saturation temperature at 26 in. is 125° F.

Latent heat at 26 in. is 1,023 Btu/lb.

Sensible heat given up by each pound of condensed steam is $125 - 90 = 35$ Btu.

Total heat given up by each pound of condensing steam is $1{,}023 + 35 = 1{,}058$ Btu.

Total heat absorbed by cooling water is $40{,}000 \times (90 - 60) = 1{,}200{,}000$ Btu/hour.

Weight of steam condensed is $\dfrac{1{,}200{,}000}{1{,}058} = 1{,}134$ lb./hour.

This assumes that the exhaust steam is dry, which is very unlikely. This figure should be multiplied by a correcting factor which should not be quite so large as the factor given in Section 254.

4. *By measuring the steam supplied by the boiler.* If the engine can be run for several fairly long periods with no other load on the boiler, then the water evaporated by the boiler is the steam used by the engine less any losses in the steam pipe. If the feed water to the boiler can be measured, all that is necessary is to see that the water level in the boiler is the same at the end as at the beginning of the test. If the feed cannot be measured, the boiler feed can be stopped and the amount by which the water level drops in the glass can be noted. A large number of readings should be taken and averaged, and the amount of water used calculated from the boiler dimensions. This method is not accurate, but it may be sufficient to draw attention to some glaringly inefficient engine.

256. THE INDICATOR. The indicator was invented by James Watt to measure the power of his engines. He knew that the pressure acting on the area of the piston was the work actually done by the steam, but realised that only part of this work was available at the engine shaft owing to losses in the engine. He carried out experiments at the London brewery of Barclay and

Perkins and found that a heavy dray horse, attached to a rope which was passed over a pulley above a deep well, could lift 100 lb. at 2·5 miles an hour. This is just 22,000 ft. lb./min. Watt added 50 per cent. to this figure in order to give his customers good measure and rated his engines on the basis of "Indicated Horse Power" of 33,000 ft. lb./min.

The indicator will tell us many interesting things. It can be used to give an idea of the actual steam consumption of a slow speed engine whose steam cannot be metered due to pulsations. It gives us the power developed by the steam inside the cylinder. It shows up many faults or good points in the design and condition of the engine. But—it must be used with care and intelligence and only an expert can interpret the indications obtained from a high speed engine.

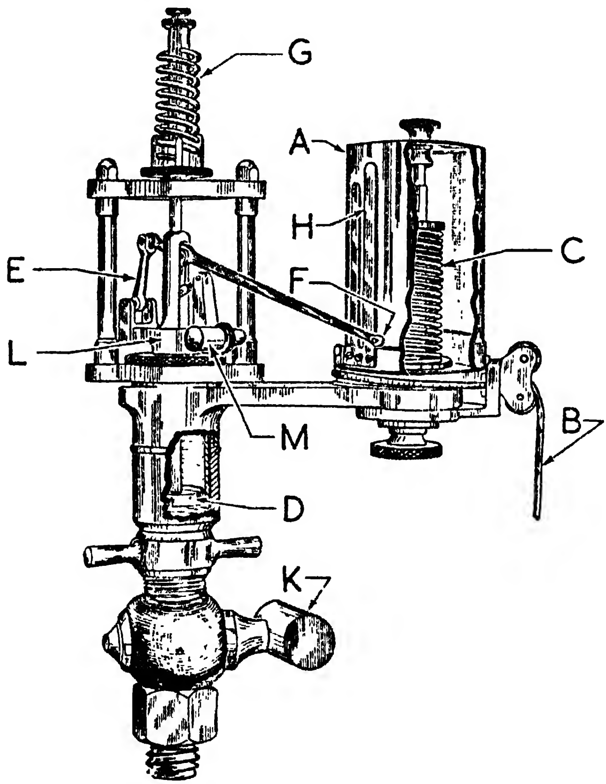


FIG. 90. THE ENGINE INDICATOR

The indicator is a small piece of mechanism which can be attached to the cylinder of the engine and from which a chart can be obtained showing the *pressure in the cylinder vertically and the position of the piston horizontally.* Fig. 90 shows one of many types of indicator. The chart is carried on a drum A, which is given a to and fro turning movement by a cord B wound round the base of the drum. The other end of the cord is attached to a suitable mechanism

driven by the engine crosshead or piston rod. A spring C inside the drum opposes the pull of the cord and keeps the cord tight. The cord is specially made for the purpose and has a wire core to prevent stretching.

257. INDICATOR RIGS. It is important that the motion of the drum should correspond as closely as possible with that of the piston. Care must therefore be taken in the design of any linkage or gear which is used for transferring the reciprocating motion of the piston to the end of the drum cord. It is always necessary to employ some method of reducing the motion, because the length of the indicator diagram is not more than about 3 in., whereas the engine stroke may be as much as 5 ft. It is relatively easy to rig up something quite suitable for attachment to an open-motion slow speed engine, but in the case of high speed totally enclosed engines it will probably be necessary to acquire from the makers of the engine the necessary gear, which they are generally in a position to supply. Some simple arrangements which give an exact reproduction of the motion of the piston on a reduced scale are shown in Fig. 91.

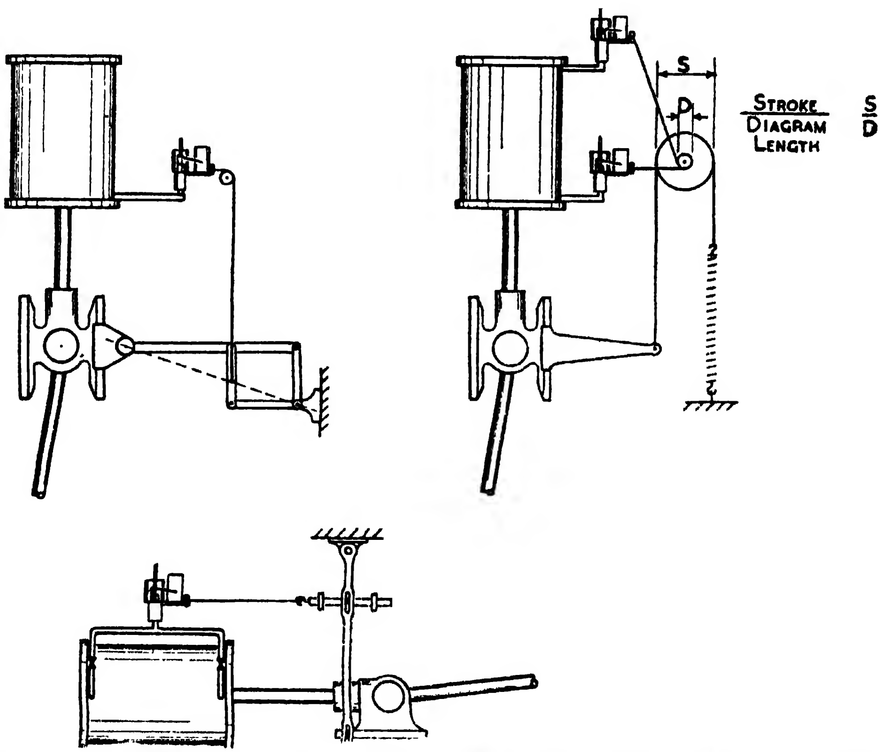


FIG. 91. INDICATOR RIGS THAT GIVE EXACT SCALE REPRODUCTION OF PISTON MOVEMENT

The indicator itself is attached to a cylinder cock, or to a special pipe leading from the end of the cylinder. A small piston D, Fig. 90, in the indicator body is connected by a straight line linkage E to a pencil F which moves vertically up and down the drum parallel with the drum's axis. The motion of the

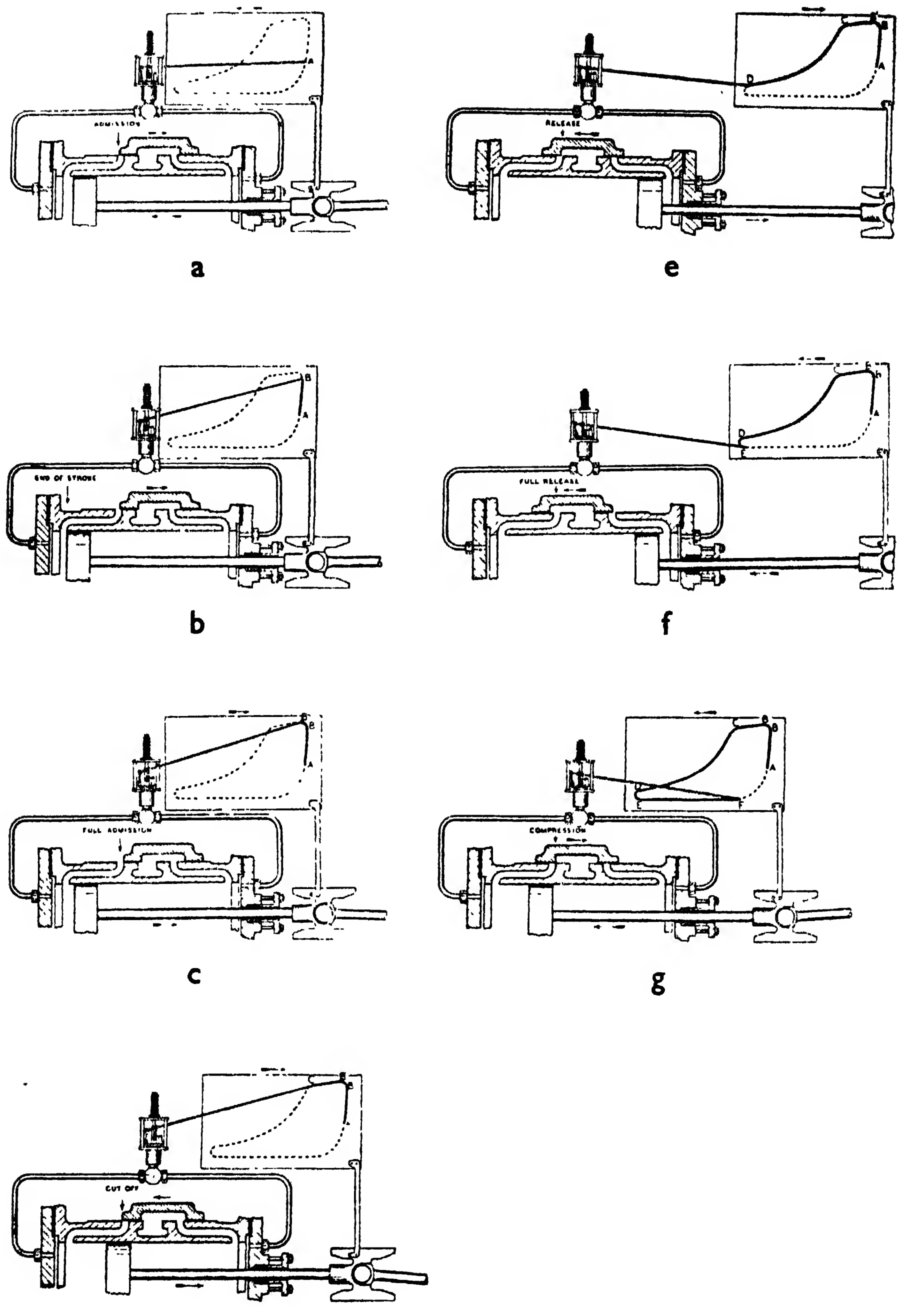


FIG. 93. THE PRODUCTION OF AN INDICATOR DIAGRAM

multiplied by the number of the spring gives the "Mean Effective Pressure" on the piston during the stroke. A force is operating over a distance, so that work is being done. If we multiply the force by the distance we get the amount of work done per stroke. If we multiply the work per stroke by the number of strokes per unit of time we get the rate at which the work is being done or the "Indicated Horse Power".

The average height of the diagram can be found in two ways, by means of a planimeter, or by the mid-ordinate method. If a planimeter is used, its readings are areas and the area of the diagram is work. So that if a number of indicator diagrams are taken regularly, the planimeter area can be calibrated for different indicator springs to give the work done per stroke direct. The mid-ordinate method is illustrated in Fig. 94. The diagram is divided into ten equal strips,

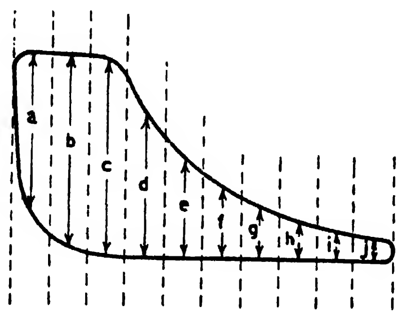


FIG. 94. FINDING THE MEAN EFFECTIVE PRESSURE BY THE MID-ORDINATE METHOD

as shown by the broken lines. The height of each strip is then measured off at its mid-point, as at a b c d, etc. The average or mean height of the diagram is then found by adding the height of the mid-ordinates and dividing by ten.

If the engine is double acting (and there are few single acting engines in use nowadays) it is necessary to take a diagram from each end of the cylinder. The two diagrams should be superimposed on the same card, when they can be compared very easily by eye.

259. THE INDICATOR DIAGRAM. Fig. 93 shows the way in which a normal indicator diagram is obtained. In these diagrams the card is shown as being fixed to the crosshead and the pencil arm of the indicator is shown of exaggerated length so as to draw a big picture. The diagram is being taken from the head, or left hand, end of the cylinder.

In Fig 93a the piston, travelling to the left, has almost reached the end of the stroke and the admission valve has just opened. The steam rushing into the cylinder raises the pressure from the admission point A to point B in Fig. 93b.

Just after the piston has started to move forward the admission valve is full open in Fig. 93c and the cylinder pressure rises to a maximum at B′.

Admission continues until cut-off at C in Fig. 93d. The pressure may drop slightly between B′ and C because the piston is running away from the steam

and the admission valve is closing slightly. The line BC is the " admission line ". At point C the steam admission is cut off, and the steam expands from C to D in Fig. 93e. The line CD is the " expansion line ".

At point D and with piston and valves as shown in Fig. 93e, the exhaust valve opens and the pressure drops from D to E, in Fig. 93f, the exhaust or back pressure. At point E the piston has just started to make its return stroke and the exhaust valve is wide open. The line EF is the " exhaust line ".

The exhaust stroke continues to the position shown in Fig. 93g, when the exhaust valve closes at point F. This point varies with the engine speed, being much earlier with high speed engines than with low speed engines.

From point F to point A, that is from the piston position shown in Fig. 93g to 93a, the valve is closed to both admission and exhaust. The steam remaining in the cylinder is compressed thus cushioning the piston and relieving the crankshaft of considerable inertia stresses.

260. TO FIND THE INDICATED HORSE POWER. Indicated horse power is the horse power as shown by the force acting on the piston indicated on the diagram. It is greater than the real horse power delivered by the shaft or flywheel owing to the mechanical losses in the engine. It is, however, not nearly so much greater as Watt assumed.

Consider a double-acting engine ; in this case there will be two diagrams similar to those shown in Fig. 101. Take the mean height of the two diagrams from either end of the cylinder and multiply this height in inches by the spring number and we get the mean effective pressure P. Call the area of the piston A. The area of the cross section of the piston rod and tail rod must be deducted from the diametral area of the piston to get at the area on which the steam is acting. A must be measured in square inches. Call L the length of the stroke in feet and N the number of strokes per minute. Then :—

$$\text{I.H.P.} = \frac{P\,L\,A\,N}{33{,}000}$$

For double acting engines N is twice the speed of the engine in revolution per minute.

261. INTERPRETATION OF INDICATOR DIAGRAMS. Indicator diagrams are useful not only for the purpose of finding the power developed by the engine, but they give other very important information, particularly on slow speed engines. On high speed engines the indicator can give very queer results and no attempt should be made, without expert guidance, to interpret high speed indicator cards.

In the case of a slow running engine the point of cut-off at C in Fig. 93 can often be ascertained from the diagram with sufficient accuracy for the purpose of estimating the steam taken into the cylinder by the method referred to in Section 254.

The height of the diagram above the atmospheric pressure line during admission shows the actual pressure of the steam in the cylinder while the admission valve is open. If the admission line B C in Fig. 93 is much below

boiler pressure it means that steam is being wastefully used—some of the useful pressure drop is being dissipated somewhere to no useful purpose. In a well-designed engine working at its most economical load with proper steam piping, the admission pressure should approach the boiler pressure to within 5 per cent. or 10 per cent. Practical experience has shown that valve adjustment, piping modifications, etc., which result in higher admission pressures will lead to greater economies in the use of steam than almost any other modification that can be made to an engine.

The indicator diagram will often reveal faults that may arise in the operation of the valves, excessive cylinder condensation, throttling, leakages, etc. It is, however, possible for two defects to occur simultaneously which will not show up on the indicator diagram—for example, valve leakage and piston leakage occurring together can mask each other. The most usual faults and some other points and their manifestations on the diagram will now be discussed.

262. DEFECTS REVEALED BY THE INDICATOR DIAGRAM. In Figs. 95 to 104 the normal diagram is shown by the full line, and the actual diagram that is being criticised is shown as a broken line.

Fig. 95. Fault outside the engine.

The broken line shows that cut-off takes place too early, in an engine in which the cut-off is controlled by the governor. This occurs when the engine is working at a load that is too low for it. The loop is caused by the steam expanding down to a pressure well below exhaust pressure. This is negative work and has a braking effect on the engine.

Remedy. Reduce the speed of the engine by using a larger driving pulley or a smaller driven pulley, or try to find some extra useful work for the engine to do.

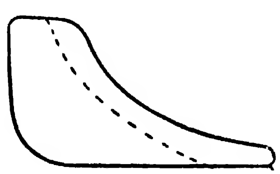


FIG. 95. CUT-OFF TOO EARLY—UNDERLOADED ENGINE

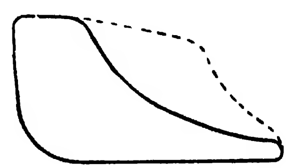


FIG. 96. CUT-OFF TOO LATE—OVERLOADED ENGINE

Fig. 96. Fault outside the engine.

The broken line shows that cut-off is taking place too late, in an engine in which the cut-off is controlled by the governor. This occurs when an engine is overloaded for its speed.

Remedy. Increase the speed of the engine if this is permissible, by using a smaller driving pulley or a larger driven pulley, or by transferring some of its load to another engine.

Fig. 97. Fault outside the engine.

The broken line shows that the engine is underloaded, in an engine which is throttle governed and where the cut-off is fixed.

Remedy. Set the valve gear to give an earlier cut-off, or decrease the speed of the engine by altering the pulley sizes.

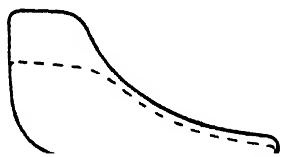


FIG. 97. UNDERLOADED THROTTLE-CONTROLLED ENGINE

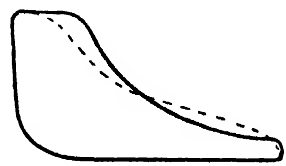


FIG. 98. EXCESSIVE CYLINDER CONDENSATION

Fig. 98. Fault outside the engine.

The broken line shows a rapid fall of pressure at first and then an increase of pressure above the normal towards the end of the stroke. This indicates excessive initial condensation in the cylinder. The flash from the large amount of condensate with the dropping pressure causes the pressure to be unduly high as the piston travels along the expansion part of the stroke.

Remedy. See notes on cylinder condensation, Sections 245 to 248.

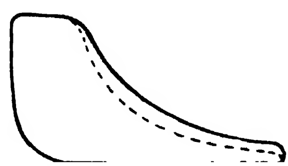


FIG. 99. PISTON OR EXHAUST VALVE LEAKAGE

Fig. 99. Fault in the engine.

The broken line shows that the pressure falls more rapidly after the point of cut-off than in the normal diagram. This indicates leakage past the piston.

Remedy. Test the cylinder bore and piston for wear and the rings for wear or breakage. If the piston, cylinder and rings appear to be in order, the leak may be through the exhaust valve. See notes on tests without the indicator, Sections 266 to 273.

Fig. 100. Fault in the engine.

The broken line shows that the pressure rises above the normal after cut-off. This indicates that the admission valve is continuing to admit steam after it is allegedly shut.

Remedy. Test the valve for leakage. See notes on tests without the indicator, Sections 266 to 273.

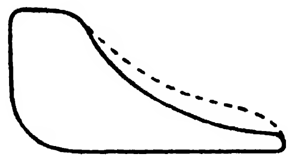


FIG. 100. ADMISSION VALVE LEAKAGE

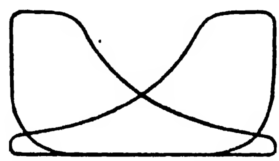


FIG. 101. ENGINE IN FIRST-CLASS ORDER

Fig. 101. No faults inside or outside engine.

This Fig. shows a pair of first-class diagrams for both the in and the out strokes on a moderate speed double-acting engine which is in good condition.

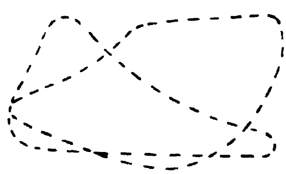


FIG. 102. VALVE SHIFTED ON SPINDLE

Fig. 102. Fault in the engine.

This pair of distorted diagrams shows that all the events occur too early at one end of the cylinder and too late at the other end. This diagram should be studied in conjunction with the distortions shown in Section 264. If the engine is fitted with a piston valve or slide valve, this diagram suggests that the valve has moved from its correct position on the spindle.

Remedy. Re-set the valve. See Section 273.

(87664)

Fig. 103. Fault in the engine.

The pair of diagrams shown in Fig. 103 would be more or less normal in a high speed engine, but in a slow speed engine such a diagram would mean that all the events occur too early at both ends of the cylinder. This suggests that the eccentric is fixed too far forward relative to the crank.

Remedy. Re-set the eccentric to give the correct valve setting. See Section 273.

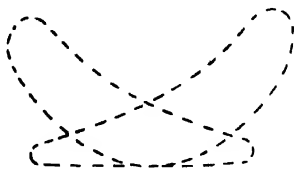


FIG. 103. ECCENTRIC TOO FAR FORWARD IN SLOW SPEED ENGINE—CORRECT FOR HIGH SPEED ENGINE

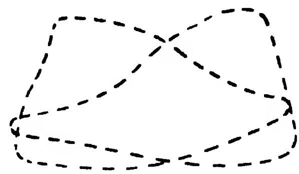


FIG. 104. ECCENTRIC TOO FAR BEHIND

Fig. 104. Fault in the engine.

This diagram shows the exact reverse of the conditions shown in Fig. 103. All the events occur too late. This indicates that the eccentric is fixed too far back relative to the crank.

Remedy. Re-set the eccentric to give a correct valve setting.

Both the foregoing diagrams should be studied in comparison with the distortions shown in Section 264.

The indicator diagram used in the foregoing examples as a standard is a good notional diagram for a moderate speed engine. Diagrams from engines in first rate condition can vary greatly in shape. Fig. 105 shows the kind of extremes. These diagrams should be compared with Figs. 109 and 117. All

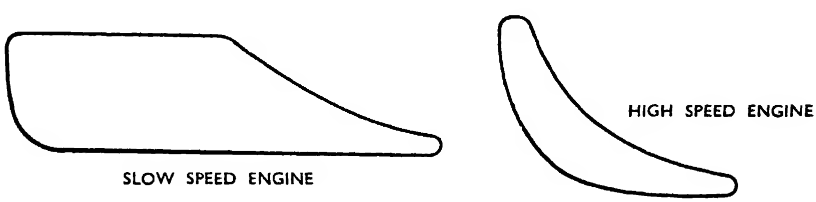


FIG. 105. TYPICAL GOOD DIAGRAMS

these diagrams are as good as can be expected from the particular engine, at the particular load, at the particular speed. It is impossible to lay down dogmatically that the diagram from an engine should have an exact particular shape. Only broad indications can be given. Adjustments should be made in each engine until the most seemingly ideal diagram is obtained.

263. POSITIVE AND NEGATIVE WORK. The line traced on an indicator diagram is a line of *pressure*. It shows the pressure acting on the piston at all positions along the stroke. From this it follows that all the area below the line on the working stroke is positive work, from which must be deducted the area below the line on the exhaust stroke. This may seem obvious, as the difference between these two areas is the area enclosed by the diagram, but we

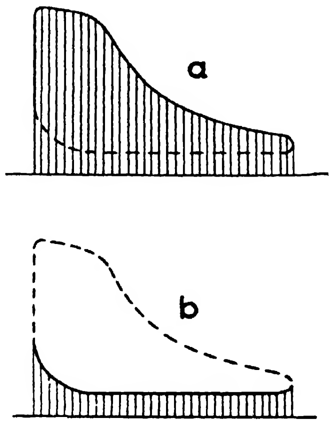


FIG. 106. POSITIVE AND NEGATIVE WORK

shall see that there can be some very queer diagrams. Fig. 106a shows the positive work done, represented by the area below the pressure or working line. Fig. 106b shows the negative pressure against which the steam has to work, on the exhaust or idle stroke. The net work is the difference between the two and of course is the area of the diagram.

264. DISTORTIONS, LOOPS AND HOOKS. The ripples shown in Fig. 107 are almost certainly caused by water in the pipe between the cylinder and the indicator.

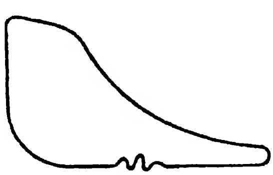


FIG. 107. WATER IN PIPE

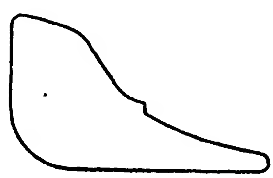


FIG. 108. GRIT OR FAULT IN MECHANISM

The step in Fig. 108 probably means grit or some other obstruction to the free movement of pencil or drum. It might be caused by a defect in the cord where it passes over the pulley.

If the kink or step is repeated elsewhere in the diagram a good idea can be obtained from its position as to whether the jerky movement is in the pencil gear or the card gear.

Section 262 dealt briefly with some typical diagrams. Certain peculiar points will now be dealt with in greater detail.

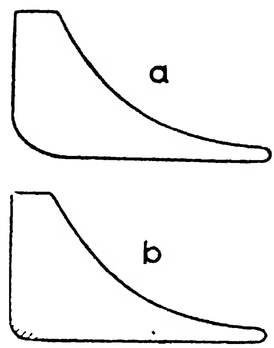


FIG. 109. INSUFFICIENT COMPRESSION

Fig. 109. No compression.

Fig. 109a shows a good normal Corliss engine diagram. Fig. 109b shows a Corliss diagram where the exhaust valve stays open too long so that there is virtually no compression. The little shaded area is, it is true, a small power gain, but the bearings may be overloaded and the engine may thump.

Fig. 110. Admission too early.

The admission line A B in Fig. 93 should be vertical—it should be vertical in all slow speed engines (except uniflow engines). If the line slopes backwards it means that the steam is being admitted too early—that the steam valve is opening too soon. This increases the negative work, as the flywheel has to drive

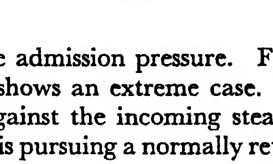


FIG. 110. ADMISSION TOO EARLY

the piston back against the admission pressure. Fig. 110a shows too early an admission, and Fig. 110b shows an extreme case. The loop is made because the piston, driving back against the incoming steam, exerts a higher pressure than can the steam when it is pursuing a normally retreating piston. The shaded area is lost work.

Fig. 111. Admission too late.

When admission occurs too late the admission line leans forward as shown in Fig. 111a. This wastes the useful power area that is shaded. With extremely late admission a loop may occur as is shown in Fig. 111b. The loop can be

due to one of two causes. During compression the steam is heated up because work is being done on it. It may now be hotter than the cylinder and will give up some heat and may partly condense. If admission does not occur promptly

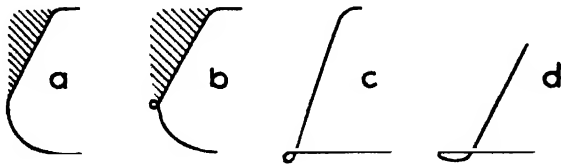


FIG. 111. ADMISSION TOO LATE

this will cause a drop in pressure and hence a loop. If the compressed steam can leak past the piston or out of a valve this will also cause a loop or increase the size of a condensation loop. Fig. 111c and 111d show two loops that occur with very late admission when compression is absent. Lack of compression can be due either to faulty valve setting or to leakage.

Fig. 112. Incorrect release.

Fig. 112a shows that release is taking place too late. Fig. 112b shows release occurring too early. The work wasted is shown by the shading.

FIG. 112. INCORRECT RELEASE

Fig. 113. Condensation.

The hook shown in Fig. 113a and more pronouncedly in Fig. 113b almost always means excessive initial condensation. As the admission line is vertical there is nothing wrong with the setting of the admission valve so that the drop in pressure must be due either to condensation or to leakage.

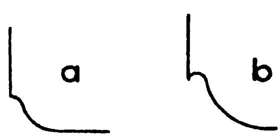


FIG. 113. EXCESSIVE INITIAL CONDENSATION

Fig. 114.

Fig. 114a shows a diagram that appears in a number of text books (on engines —not figure-skating). What is the area enclosed by the diagram? This is where the positive and negative work comes in. Fig. 114b shows the positive

work below the line that is drawn during the working stroke. Fig. 114c shows the negative work on the exhaust stroke. These areas are almost identical. How then can the engine run ? The author had always thought that the writers

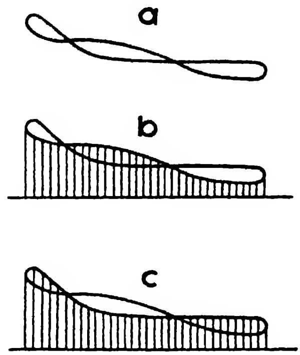


FIG. 114.

of the text books were, in Air Force parlance, " shooting a line " when they showed this diagram, until he received from a friend an exact replica, which solved the mystery. The friendly diagram was taken from the low pressure cylinder of a back pressure compound engine on half load. Clearly such a cylinder was cutting no ice and was just a passenger.

265. VALVE GEAR CHARACTERISTICS. The indicator shows up very nicely the good and bad points of different valve gears, from the thermodynamical point of view ; it does not necessarily show up mechanical trouble and heavy maintenance.

Fig. 115 shows the diagrams given by an engine on various loads with an ordinary slide valve and plain eccentric gear, with throttle governing. At anything below full load the steam is wastefully reduced in pressure and a large amount of this low pressure steam has to be admitted to the cylinder.

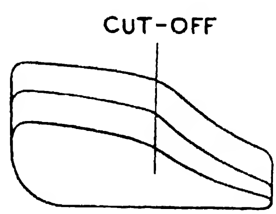


FIG. 115. FIXED CUT-OFF, THROTTLE-GOVERNED SLIDE VALVE ENGINE

Fig. 116a shows the enormous improvement obtained with the same type of slide valve and eccentric but arranged for cut-off governing. The steam is used at its top pressure and much less steam has consequently to be admitted for any given load below full load.

Fig. 116b shows the additional improvement obtained from trip valves. The cut-off is crisp and clean, instead of indeterminate and slovenly. It is difficult to determine the exact point of cut-off from a slide valve diagram as can be clearly seen in Figs. 115 and 116a.

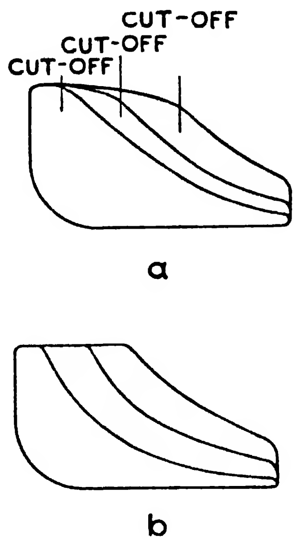


FIG. 116. CUT-OFF-GOVERNED ENGINE WITH SLIDE AND TRIP VALVES

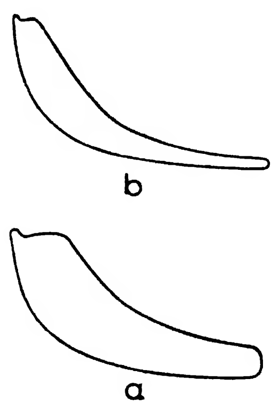


FIG. 117. UNIFLOW ENGINE

Fig. 117 shows the sort of diagram recorded by a Uniflow engine on about half and on full load. The virtues of the Uniflow are not really manifest from the indicator diagram. Most of the benefit that accrues from the Uniflow engine is the small amount of cylinder condensation, and consequently a greater amount of expansion can be done in one cylinder.

Fig. 118 shows indicator diagrams of a winding engine taken by B. L. Metcalf from the first revolution at full admission to the nineteenth revolution when the steam was shut off. The characteristics of these indicator diagrams can be compared with the various diagrams that have just been considered, and the comparison will be found very interesting.

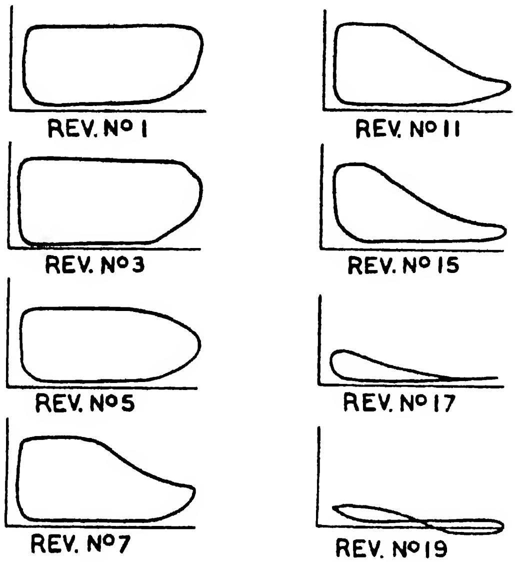


FIG. 118. WINDING ENGINE DIAGRAMS

266. TESTS WITHOUT THE INDICATOR. It is often possible to get valuable information about the state of the engine without the use of an indicator, or to confirm defects whose existence are suggested by the shape of the indicator diagram. Much depends on the type of engine. Simple engines are easier to test than compound engines. Slow speed engines generally lend themselves more readily to such tests than high speed engines.

The information that is wanted is of three kinds : whether the valves leak ; whether the piston leaks ; whether the setting of the valves is the best that can be made. The tests are of two kinds : by measurement and inspection ; by the use of steam. Measurement is simple and straightforward, but a thorough inspection may necessitate the engine's being out of commission for some time. The use of steam is more tricky but is in many ways more satisfactory and gives more information (except to the very inexperienced) than measurement and inspection.

Piston and valve leakage are common—far too common and far too serious. Engines should never be allowed to run for years without tests, which should be made at regular intervals, say six-monthly. Engines often run for long periods with broken piston rings which are unsuspected causes of extravagance and can be detected by the simple tests described below.

267. MEASUREMENT AND INSPECTION. Valve covers should be removed and slide valve bedding tested with feelers or red lead, the valve seat face being tested with a straight edge. Piston valves can sometimes be tested by removing the covers and using long feelers. Cylinder covers should be taken off and the cylinder bores should be gauged at many points on the vertical and horizontal diameter all along the stroke to see whether the cylinder has become oval or barrel-shaped. It is necessary to draw the pistons to find out whether the piston rings are broken.

268. STEAM TESTS. It is sometimes inconvenient to carry out the measurements and tests described in Section 267. It may be impossible to put the engine out of commission. Even if the measurements and gaugings are carried out, a decision as to the amount of wear shown by feelers and gauges can only be taken by someone of considerable experience. Feelers will not detect broken rings and there may not be an opportunity to lay the engine off for the time needed to draw the pistons. By using steam the amount of leakage can be seen, but the steam tests must be done with care and intelligence, and must be carried out with the engine as hot as possible.

It is difficult to apply the steam tests to the low or intermediate pressure cylinders of compound or multiple expansion engines. If, however, leakage is found on the high pressure cylinder, it is not unlikely that the other cylinders are leaking too. Anyhow, the engine should be laid off as soon as possible for repair to the high pressure end. If inspection shows that the low pressure cylinders are in the same condition as the high pressure end, then the same repairs will probably be needed by the low pressure parts.

269. ADMISSION VALVE LEAKAGE. The engine must be set in such a position that the valves are shut, so that no steam should enter either end of the cylinder. (The high pressure cylinder only of a compound or multiple expansion engine is being considered.) The method of doing this varies with different types of engine.

(*a*) Corliss or drop valve. The movement of these valves can be seen and there is no difficulty in setting the engine so that the valves are in the shut position.

(*b*) Reversing engines. Set the valve gear in the mid position

(*c*) Plain piston valves. Set the engine so that the valve spindle is in the mid position. Find this position by marking the valve spindle at its extreme points of travel and taking the half-way point.

(*d*) Plain slide valves. As there may be lost motion between the spindle and the valve, it is best to remove the valve cover and find the two points where the valve is brought to the mid point. Mark these points on the flywheel. Replace the cover.

(*e*) Engines with more than one high pressure cylinder. Blank off the steam inlet to all cylinders except one by means of a slip joint, or by removing a branch and fitting a blind flange, or by any other suitable means. Then apply the tests to each high pressure cylinder in turn taking the appropriate steps as in a, b, c or d above.

Under c above the procedure assumes correct valve setting. If there is any doubt about the valve setting, it should first be checked as explained in Section 273 before doing the leakage tests.

Having removed the tail pipes from the cylinder drain cocks so that the cocks blow into the engine room, set the engine so that the valves are shut. Open the cylinder drain cocks. Open the steam stop valve cautiously. If any steam comes out of the drain cocks it must have come through a leaky valve. If the valve is leaking very seriously it may allow sufficient steam to enter the cylinder to move the engine, though this is very unlikely because when the valves are shut the crank is almost on dead centre. Any leaks found by this test are leaks of live steam past the steam lap edge of the valve.

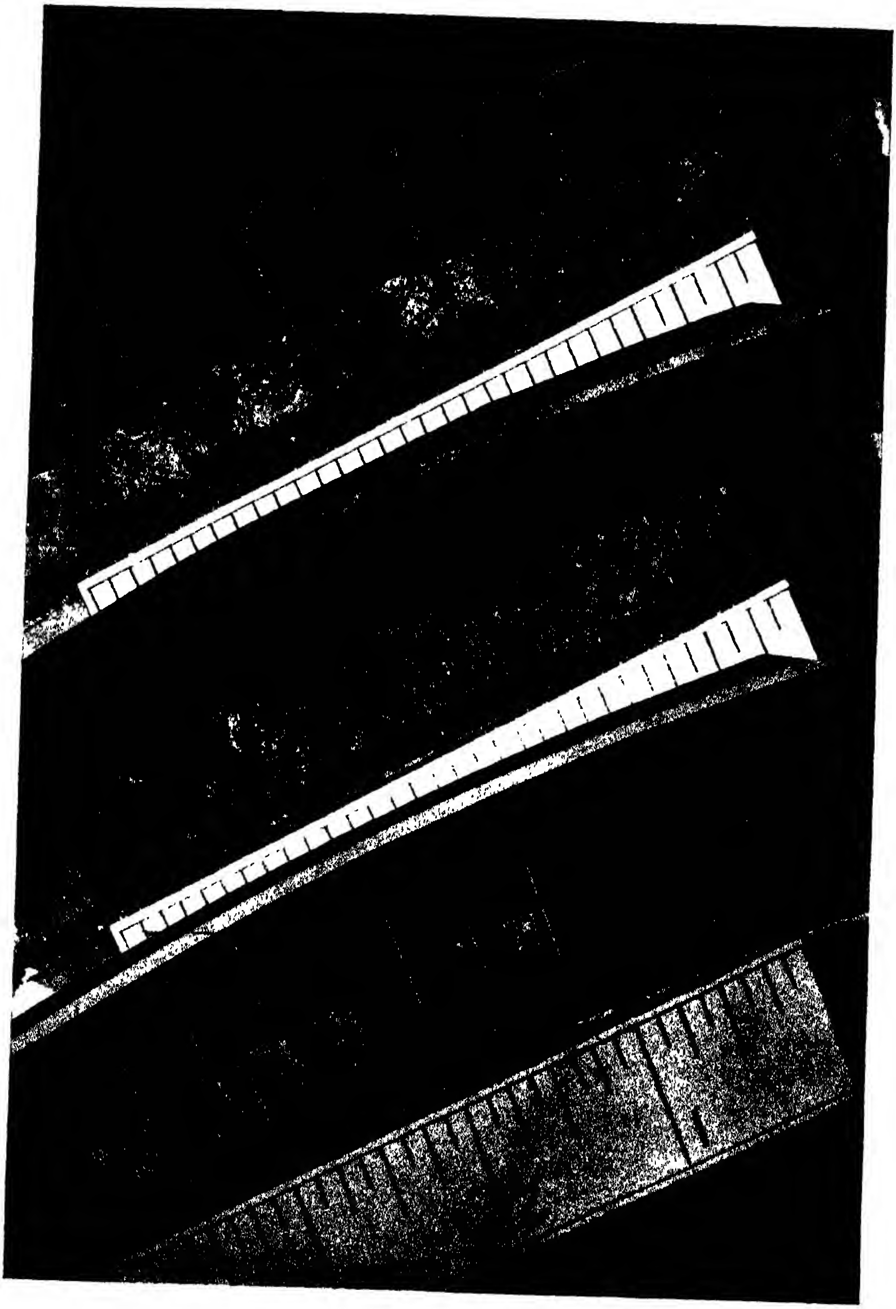
270. HIGH PRESSURE TO EXHAUST LEAKS. With flat slide valves it is possible for steam to leak through the side edge. This will allow steam to pass straight from the valve chest to the exhaust port without going into the cylinder. Set the engine, as explained in Section 269, in the closed valve position. Break an exhaust pipe joint as close to the cylinder as possible. Open the steam stop valve. Any steam escaping from the exhaust port must be leakage.

271. EXHAUST VALVE OR PISTON LEAKAGE. Break an exhaust pipe joint as near to the cylinder as possible. Set the engine so that the crank is 1° or 2° past the dead centre. The inlet or admission valve should now be definitely open. Admit steam to the engine. In this position the exhaust valve on the live end of the cylinder should be closed. If steam comes through and out of the broken exhaust pipe on the dead end of the cylinder, either the exhaust valve on the live end is leaking or the piston is leaking.

272. PISTON LEAKAGE. If any leakage into the exhaust is shown in the test just described in Section 271, valve leakage must be distinguished from piston leakage. Set the engine at various points, between admission and cut-off, along the stroke and open the cylinder drain cock on the exhaust side of the piston. Open the stop valve quickly and sharply. The engine will at once start, but during the fraction of time before the steam enters the other end of the cylinder, watch the open drain cock for blows. If steam blows through the cock before the piston reaches the end of the stroke it is almost certain that this steam has blown past the piston.

It is advisable to carry out this test even if the test in Section 271 has not shown leaks because the cylinder may be worn in some parts and not at the end of the stroke.

The foregoing tests are not entirely conclusive and they can be greatly improved by jamming the engine so that it cannot move. This allows a higher pressure to act on possible leaking parts and gives plenty of time for observation to be made. Jamming the engine can, however, be dangerous, and under no circumstances should it be done without every possible care being taken that the jamming " weapon " cannot come adrift. Where the fly-wheel has spokes and is exposed the engine can be most easily jammed by a baulk of timber passed between the spokes and so wedged that the engine is fixed in the correct position as already described. If the fly-wheel has a solid web it may be possible



TOP: Turbine blades which maintained specification performance in spite of extensive damage.

MIDDLE: Same blades 18 months later when performance had perceptibly dropped although there is little extra visible deterioration.

BOTTOM: Rotor re-bladed.

(All full size: see Section 274.)

to wedge a bar in one of the barring holes, but great care must be taken that the jamming bar is really tightly wedged in.

Having carried out these tests on the high pressure cylinder and thus knowing its condition, some idea as to the state of the low or intermediate pressure cylinders can be obtained by running the engine slowly, under load if possible, under steam with all the cylinder drains open. By comparing the blows on the low and intermediate cylinders with the blows on the high pressure cylinder, making mental reservation for the difference in pressures, some sort of an idea may be obtained as to the state of the whole engine.

273. VALVE SETTING. The point of admission and the point of cut-off can be found by turning the engine degree by degree with the cylinder drain cock open on the live end and then opening the steam stop valve after each movement of the engine. As soon as steam appears at the cock (assuming that there is no serious leakage) this is the point of admission, which should be just before dead centre—2° to 3° before with a slow speed engine, up to 8° or 10° with a high speed engine. When the engine has been turned to such a position that no steam appears at the cock when the stop valve is opened, this gives the point of cut-off.

274. STEAM TURBINES. The actual steam consumption can easily be found from a steam meter. The actual load on a turbine, unless it be an electrical load, is just as difficult to estimate as the load on a reciprocator. The methods described in Chapter 2 can be used to find the cycle efficiency and the probable efficiency ratio.

The steam turbine can and does deteriorate. Blades get eroded and corroded. Blades sometimes get coated with scale from boiler carry-over. Glands, both shaft and interstage, can leak unduly. There are simple instrument readings which can give quite a fair idea as to the goings on inside a turbine.

The pressures at all possible points in the turbine should be read at a moment when the load (if this is known) is so and so, or at a particular governor opening. A scale should always be attached to the governor rod. Then at a given load, with standard inlet pressure, the governor opening can be noted. If it is found that the governor opening is increasing for a particular load, it means that the steam consumption is increasing or that the steam is having difficulty in passing through the blading. If the load is not known or even if it is, a particular governor opening can be compared with pressure in the nozzle chest, in the first wheel chamber and with the exhaust pressure. Fig. 119 shows the pressure readings of a 5,000 kW back pressure turbine in the first wheel chamber at constant load and constant inlet pressure over a period of 25 days. This gradual rise of chamber pressure was interpreted (correctly !) as meaning that the reaction blading in the low pressure end of the turbine was building up a deposit. An increase of pressure drop across any particular part of a turbine means that the steam is finding difficulty in getting through. This generally means that the steam passages are getting constricted due to deposit or scale. Erosion or wear will increase the areas for the steam to pass through and will therefore tend to cause a reduction in the pressure drop across the worn blading.

A constant daily or weekly log of pressure readings, quantity readings and any other data should be kept and plotted on a chart. It is difficult to see from a row of figures that some slow change is occurring, but this is at once visible on a chart, whose scales should be small so as to smooth out chance local variation and not distract the eye.

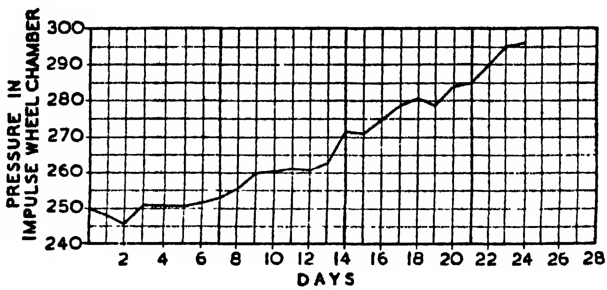


FIG. 119. EFFECT OF DEPOSIT ON TURBINE BLADING

A turbine is often spoken of with reverence as a reliable machine which needs hardly any maintenance. This is true up to a point. A turbine, once it has got over its growing pains, will run with remarkable reliability and with the minimum of service for years, but when it does need repairing it makes up for lost time. Reblading and reglanding are most expensive luxuries. Reglanding pays a good dividend. It saves steam and increases the power output. Reblading should generally be postponed until it is really necessary. Because blades are pitted, eroded, chipped and just look fit for the scrap heap does not necessarily mean that reblading must be done. Pages of high grade mathematics are used for calculating the exact shape of blade, but blades that look like chunks of lava seem often to work just as well as when new. Provided there is sufficient metal for strength—and, most important, provided the shroud ring is in good condition—it is quite remarkable how well turbines will perform in spite of most moth-eaten blades. The frontispiece shows some of the blades on the first wheel of a turbine in the author's factory. Six months after the top photograph was taken this turbine, in this condition, was using exactly the steam/kWh tendered by the maker (apart from loss due to damaged glands).

Turbines should not be left unexamined for years. Serious weakening may go undetected and the first sign of trouble is a frightful smash. It is a good rule to open up a turbine for inspection every two years unless the steam quality is completely above suspicion. If there is any doubt as to the steam quality the turbine should be examined yearly.

This book is no place for a description of the theory of turbines, but one point can be mentioned. An impulse turbine is a windmill ; a reaction turbine is a catherine wheel. Obviously the first wheel on a turbine must be chiefly an impulse wheel. Reaction is generally more efficient than impulse, and its superiority increases with lowered pressures. For this reason the low pressure end of a turbine exhausting to condenser is almost always fitted with reaction blading, but many back pressure turbines have nothing but impulse stages.

It has been found that deposits due to dirty or oily steam form more readily on reaction blading than on impulse blading. Impulse blading may therefore be preferred for exhaust or mixed pressure turbines taking exhaust steam from reciprocating engines.

275. THE IMPORTANCE OF VACUUM. In Chapter 2 it was shown how large a gain is obtainable by increasing the vacuum at which an engine or turbine exhausts. It was also explained that the effect of vacuum is much more important in turbines than in reciprocators.

Here is a little tabulation constructed from Table X in Section 104.

INITIAL PRESSURE	ESTIMATED HORSE POWER FROM 10,000 LB. STEAM/HOUR			
	ENGINE		TURBINE	
	20-IN. VAC.	15-IN. VAC.	20-IN. VAC.	15-IN. VAC.
100 psi.g.	453	441	573	471
150 „	506	490	660	507
200 „	541	521	633	549

It is therefore of great importance that every effort be made to keep up the vacuum. B.E.A. Vigers has enunciated a "profound thermodynamical truth", 'The vacuum that is obtained on any plant depends on the leaks that are permitted.' Every leak must be found and stopped. The plant should be examined at every joint with a lighted taper, which will show up leaks clearly.

* * *

CHAPTER 8

TRAPS

These but the trappings and the suits of woe.
SHAKESPEARE. *Hamlet.* 1601.

MOST factory managers and engineers would agree with Hamlet as they regard traps as unmitigated nuisances. This is almost entirely the fault of the factory maintenance staff or the fault of those responsible for installing the traps. There are many kinds of trap. Certain kinds are quite unsuitable for certain situations, but to many people a trap is just a trap. This Chapter will not deal exhaustively with traps and trapping, because the subject is dealt with, with a wealth of most readable detail, in "Steam Trapping and Air Venting" by L. G. Northcroft, to which the reader is referred, see Section 805. The application of traps and some modifications to them are dealt with in the next Chapter.

276. WHAT A TRAP IS AND DOES. A steam trap is a device which distinguishes between water and steam and automatically opens a valve to allow water to pass out but which closes to steam and traps it. Traps are of two broad kinds. Those which distinguish water from steam owing to the difference in density of the two—these are the mechanical traps ; and those which distinguish by means of temperature—these are the thermostatic or expansion traps. This at once gives some clue to the different characteristics and usefulnesses of the two types.

Saturated steam and the condensate it is forming have the same temperature. A steam trap which makes its choice by temperature must impose a delay on removing the condensate, because the condensate must cool to below steam temperature before the trap can make up its mind that it must open. On the face of it this seems to put the thermostatic traps at a great disadvantage. This is only true up to a point. Thermostatic traps have certain advantages for certain applications which will appear in subsequent sections.

277. FLOAT TRAPS. There are many designs. All depend on the fact that a float will float in water and sink in steam. When condensate enters the trap the float rises and opens the discharge valve. The steam pressure then blows the condensate out. The float sinks, or fails to rise further, and a position of equilibrium is reached where the condensate is discharged at the same rate as it is flowing into the trap.

Such a trap is shown in Fig. 120. (None of the traps here illustrated are intended to represent commercial designs. They are figments of the author's.) The float lever A is connected to the valve arm B by means of the toggle link C. The three levers are so arranged as to give a small movement to the valve for a large movement of the float when the trap is nearly empty, whereas the fuller it gets and the higher the float rises the more rapidly does the valve open.

It will be seen that the whole mechanism is accessible without interfering with the assembly of the trap to the piping, and that the valve and its seat can be renewed by simply removing the casing. This would seem at first sight to

be an admirable design feature for any trap. There is, however, something to be said for those designs in which the trap must be removed from the pipe line before the trap can be opened up. This point is discussed in Section 284.

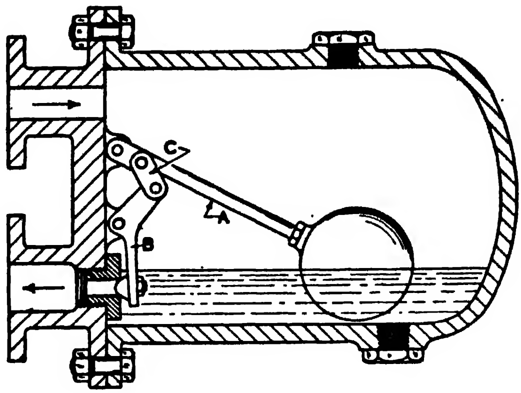


FIG. 120. PLAIN FLOAT TRAP

278. LIMITED DISCHARGE CAPACITY OF DIRECT-ACTING TRAPS. There are certain limitations to this simple float trap as there are to all direct-acting mechanical traps. The valve must be small, or the float will not have sufficient buoyancy force to open it. If the area of the valve were ·5 sq. in. and if there were a pressure inside the trap of 40 psi.g. the float would have to exert a buoyancy force of 20 pounds on the valve to open it. In Fig. 120 the float lever has a leverage between float and toggle pin of about 8 to 1. This reduces the buoyancy force required of the float to 2·5 pounds. The toggle has a mechanical advantage of about 3 to 1, so that a buoyancy force of less than 1 pound acting on the float will open the valve. Now 1 cu. ft. of water weighs 62·3 lb. Therefore a displacement of ·016 cu. ft. or 27·6 cu. in. will exert a buoyancy force of 1 pound. The volume of a 5 in. ball is 65·6 cu. in. If we assume that the weight of the float and lever is such that the float floats half submerged, complete immersion of a 5 in. float will give slightly more than 1 pound of lift. If such a trap were fitted to a steam system working at 60 psi.g. the buoyancy would never be sufficient to open the valve—the trap could not work. Nevertheless the trap would be blamed instead of the clever engineer who fitted a trap that could never possibly work.

There is therefore a definite limit to the size of the valve seat on a float trap. The buoyancy force of the float must be sufficient to open the valve, and the higher the working pressure the larger must the float be or the smaller the valve. The float cannot be increased in size with impunity ; large floats are liable to collapse. If they are made strong enough to resist collapse they become heavy and lose their buoyancy. There is therefore a definite limit to the valve size and discharge capacity of any direct-acting float trap. The buoyancy relative to the valve area fixes the maximum pressure at which the trap can work, and the valve area and the pressure drop across the valve fixes the maximum discharge capacity.

In some traps, where the valve seat is insufficiently hardened, the constant hammering of the valve so enlarges the seat that the buoyancy force of the float is insufficient to open the valve against the steam pressure. At a certain point in its life such a soft-valve-seated trap will just lock shut.

279. TRIP ACTION FLOAT TRAPS. The direct-acting float trap illustrated in Fig. 120 will discharge condensate continuously at the rate at which it is reaching the trap. Every particular float level corresponds to a particular valve opening which corresponds to a particular discharge rate at a particular pressure. This means that at very low outputs the valve will be cracked open. Valves do not stand up well to cracked-open working. Although theoretically a plain float trap should always operate continuously, in practice it often operates intermittently on very small loads. This is due to the instability of the valve at very small openings. The least disturbance may cause the valve to shut and before it can open again the full buoyancy force of the float is needed. This intermittent action of a float trap can only take place at low outputs and is beneficial as it protects the valve and seat from cracked-open working. For conditions where the trap may be working on very low flow rates, yet may occasionally have to pass large amounts of condensate, there are traps in which the float, when it has risen a considerable distance, trips the valve open. The trap then discharges full bore, the float drops and trips the valve shut.

Fig. 121 shows a trip action float trap. Fig. 121a shows the general arrangement and Figs. 121b, c and d, show the mechanism immediately before or after tripping. The float lever A, the quadrant B and the valve bell-crank C are all pivoted independently on the pin D. Pivoted at E are the two weighted pawls F and G. In Fig. 121a the trap has just emptied and tripped into the shut position. The valve K is held shut by the weight on the bell-crank C, and by the steam pressure. The quadrant B is held in the position shown by the pawl F. In Fig. 121b the trap has filled with condensate and the float has risen so that the float lever A is bearing against the weighted end of pawl F. In Fig. 121c the rising float lever A has tripped the pawl F out of engagement with the quadrant B. The quadrant weight W drops and the quadrant pin H knocks the valve open. The steam pressure blows the condensate out of the trap and the float drops. Pin L on the float lever A engages the bottom of the quadrant slot and lifts the quadrant weight W to the position shown in Fig. 121d. The quadrant weight W is now bearing against the weighted end of pawl G. In Fig. 121a the float has dropped sufficiently to cause the quadrant weight to lift pawl G and allow valve bell-crank C to shut due to the weight on the bell-crank. The cycle is now ready to start again.

One of the advantages of this type of trap (or in fact any trap with an intermittent discharge) is that it can be heard operating. The fact that it is known to be working does not necessarily mean that it is not leaking and perhaps blowing steam, but it does mean that it is getting rid of the condensate. The knowledge that the trap is working is of great importance, because traps are so often distrusted and byepassed on the least provocation. The disadvantage of intermittent discharge is that the flash steam produced from the condensate appears in short sharp puffs which may be difficult to collect and use. The noise made by an intermittently discharging trap may sometimes be disadvantageous, e.g., in a hospital.

The principal application of trip action float traps is where the output to be handled fluctuates widely, and where the plant may be tilted or swung about, as on board ship. The operation of direct-acting float traps, or of the bucket traps about to be described, may be upset by movement. The trip action float trap is well suited to ship board conditions because the valve must either be wide open or tight shut—it cannot vacillate. These traps are, however, expensive and have a lot of mechanism that can wear and go wrong. Their maintenance is rather heavy.

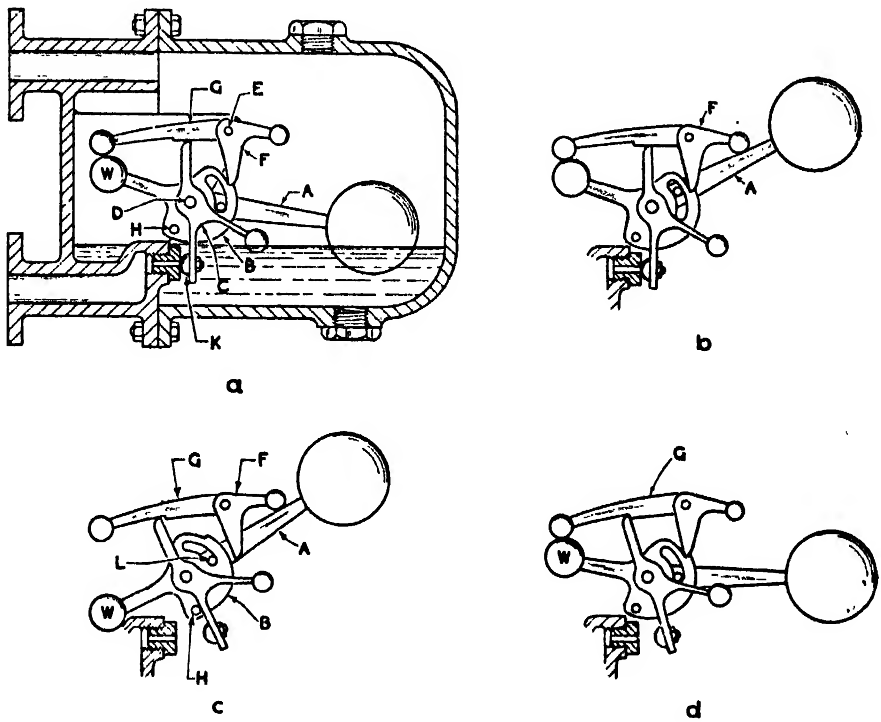


FIG. 121. TRIP ACTION FLOAT TRAP

280. BUOYANCY OF FLOATS. In some types of trap the float is heavier than water ; it would, in fact, not float. Actually the weight of the float does not matter ; it can be counterbalanced, by weight or by spring. The buoyancy force acting on the float is quite independant of the weight of the float ; the force depends only on the volume of the float, its amount of immersion and the density of the displaced liquid. The buoyancy force is equal to the weight of liquid displaced by the float. Suppose we have a float whose volume is equal in capacity to one gallon. The amount of water that this float can displace is one gallon, and one gallon of water weighs 10 pounds. The buoyancy force acting on this float when completely submerged will be 10 pounds. If the float is made of sheet copper it may weigh 3 pounds. In that case the force acting on it when it is submerged will be the weight of its own volume of water—

10 lb.—minus its own weight—3 lb. So that the net lift will be 7 pounds. If we made our float the same size but of solid lead it would weigh 114 lb. If it was counterbalanced with a weight of 111 lb. it would have a net weight of 3 pounds and the net buoyancy effect would still be 7 pounds, so that it would work just as well as the light hollow float, except that the whole thing would be nearly 80 times as heavy and therefore be more cumbersome and less sensitive. The great advantage of using a solid float is that it cannot puncture or collapse. When a hollow float gets water inside it the balance of forces is upset. It still has the same buoyancy force acting on it but it needs counterbalancing. The most reliable kind of float is a solid float made of very light material so that the whole apparatus connected to it can be light and sensitive. Some traps have floats made of solid aluminium. A solid aluminium float whose volume is such as to displace a gallon weighs only 26 lb. (See Section 292.)

281. WATER DENSITY VARIATION. Water, like other things, expands as it gets hotter. Cold water weighs 10 lb. per gallon ; water at 316° F., the temperature of 70 psi.g. steam, only weighs 9½ lb. per gallon ; water at a temperature of 497° F., the temperature of 650 psi.g. steam, weighs 8 lb. per gallon. So this reduction in the density of water imposes yet another limitation on the use of float traps for high pressure. The float must be extra buoyant to work correctly in the " light " high temperature water.

282. AIR DISCHARGE FROM FLOAT TRAPS. No float trap, unless provided with a special device, will discharge air or other incondensible gas. At the start-up of any plant all the inside of the heating surface is full of air, and this air must be cleared out before full use can be made of the heating surface. Air and/or CO_2 is present in even the best steam and finds its way eventually into the trap unless means are provided to remove it. When the trap gets full of air the condensate is unable to get in and the trap goes out of action. The trap is blamed and byepassed, when the fault lies with the plant designer who has made inadequate provision for the removal of air. The removal of air is such a big and important subject that it will have a chapter to itself, and the removal of air will only be mentioned in passing when discussing the characteristics of various kinds of trap.

283. OPEN BUCKET TRAPS. Fig. 122 shows an open bucket trap—there are many designs, but all work on the same principle. Condensate enters at A. It fills up the body of the trap. The bucket B, pivoted at C, floats in the rising condensate until the valve D closes on the seating E. The buoyancy of the bucket clearly provides the valve-closing force in this type of trap. When the valve is closed, the bucket can rise no more, so the rising condensate eventually reaches the level of the bucket top and runs over into the bucket. When sufficient water has got into the bucket to overcome its buoyancy and the force of the steam that is holding the valve shut, the bucket drops and opens the valve. The steam pressure blows the condensate out of the bucket up and out of the discharge pipe until the bucket is sufficiently buoyant to float again. The bucket then rises to the position shown in Fig. 122 and closes the valve ; the cycle then recommences. The buoyancy of the bucket is so arranged that the bucket never empties completely thus maintaining a water seal between the valve pipe and the bucket so that steam cannot blow through. The action

of an open bucket trap must be intermittent. This type of trap is therefore not suitable for application where it is important that there should be a continuous discharge.

The closing force on this trap is the buoyancy of the bucket. The opening force is the weight of the bucket. When the bucket is full, buoyancy disappears and the whole of the weight of the bucket is available to open the valve against the steam pressure holding the valve shut. It is not so easy with this type of trap to arrange for large leverage between bucket movement and valve movement. This means that if the trap is to handle large outputs it must be very large.

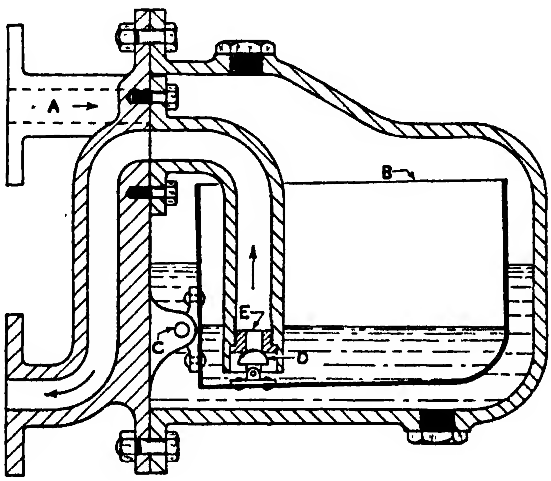


FIG. 122. OPEN BUCKET TRAP

The open bucket trap has the following characteristics. There is no possibility of the bucket collapsing. There is less chance of the bucket leaking than there is of a float leaking. Unless the leak in the bucket is quite large the trap will continue to work. If however the leak gets very large the bucket may not rise to close the valve. If a leak is present the trap will operate more frequently than when in normal condition. If, on normal load, the period of the trap action is noted and it is subsequently seen to be working on a shorter cycle, a leaking bucket should be suspected. The frequency of operation should always be observed at some constant pressure. A lower pressure will allow the trap to operate more frequently as the force holding the valve shut is smaller and can consequently be overcome by less load in the bucket. The bucket cannot collapse and there is consequently no limit to the pressure at which an open bucket trap can work except that imposed by valve-area-times-pressure and bucket weight. The bucket trap is more robust than the float trap. The fact that the discharge is intermittent has the advantage that it can be heard operating, but flash steam is more difficult to collect and use.

In both the float and open bucket traps two forces are used ; the buoyancy of the buoyant member and the weight of the buoyant member. Ruggedness and reliability demand that the weight be large and the buoyancy small. Now

in a float trap it is the buoyancy that opens the valve and the weight that closes it. In an open bucket trap it is the weight that opens the valve and the buoyancy that closes it. This possibly gives the bucket trap an advantage, but just like the float trap there is a pressure above which any bucket trap cannot work. Take Fig. 122. If the valve has an area of ·25 sq. in. and the mechanical advantage of the bucket on the valve is 2 to 1 it means that at a pressure of 8 times the weight of the bucket, the bucket will not be heavy enough to open the valve. At 120 psi.g. there will be a pressure of 30 lb. on the valve. If the mechanical advantage of the bucket is 2, the weight of the bucket necessary to open the valve will be 15 lb. If the bucket weighs only 12 lb. the trap cannot work at 120 psi.

Before a bucket trap is set to work it should be filled with water to ensure that the bucket floats, and to seal the outlet.

Like float traps, open bucket traps can get locked with air unless some means are taken to remove any air or other incondensible gas.

284. INVERTED BUCKET TRAPS. Fig. 123 shows an inverted bucket trap. Condensate enters through the lower connection and through the upturned pipe A that projects into the inside of the inverted bucket. Any air that is trapped inside the bucket B escapes through the small leak hole C. The condensate accumulates inside the trap body and inside the bucket. Provided the condensate is entering fairly fast the leak hole will be inadequate to even out the pressure inside and outside the bucket so that the water level will rise faster outside the bucket than inside. This gives the float buoyancy and keeps the valve shut. As the condensate accumulates, it fills up outside the bucket and, as fast as the leak will allow, the water rises inside the bucket. When sufficient water has risen inside the bucket, its buoyancy disappears and the weight of the bucket opens the valve. The steam pressure then blows the water out from inside the bucket and round outside and through the outlet. As soon as sufficient water has been blown out of the bucket, its buoyancy is restored, it rises and closes the valve, leaving the trap in the position shown in Fig. 123. Now it is clear that the bucket would retain its buoyancy unless the steam and/or air inside the bucket could escape. Air escapes through the leak hole and is eventually discharged with the condensate. Steam escapes through the leak and is condensed in the body of the trap. During discharge steam blows straight through the leak hole out of the discharge, but the quantity so lost is so small as to be of little importance.

The valve opening force is the weight of the bucket, as in the open bucket type. The valve closing force is the buoyancy of the bucket.

The inverted bucket trap has several very real advantages. It is very robust, simple and reliable. It will operate under conditions of movement, for example on ship board. It can be made extremely small and light—see Fig. 124. It will vent air quite sufficiently to ensure that plant never gets air locked, though it may not remove air sufficiently quickly to be satisfactory on plant working intermittently on very short cycles, for example some hemispherical boiling pans.

A trap such as is shown in Fig. 124 cannot be opened up without taking it out of the pipe line. Small inverted bucket traps are made so that the cover

can be removed without disturbing the piping. This however calls for small tortuous passages cored in the trap body, and the blockage of these passages may be more tiresome than the removal of the trap from the line.

The inverted bucket trap has some disadvantages. It must always waste a little steam. During discharge the bucket leak is discharging steam to its full capacity. During water accumulation steam must be allowed to condense in the top of the trap in order to allow the bucket to vent air and thus permit it to fill with water. If the bucket vent gets blocked, the trap locks shut.

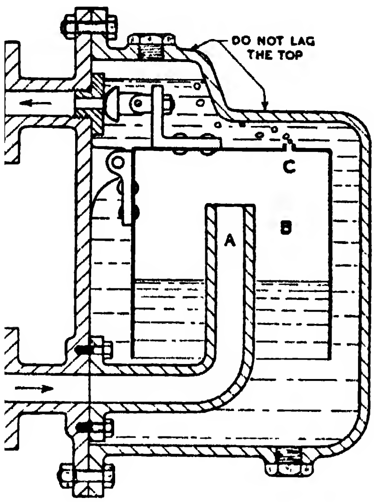


FIG. 123. INVERTED BUCKET TRAP

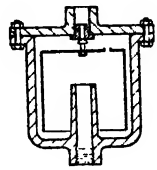


FIG. 124. SMALL INVERTED BUCKET TRAP

It is possible, under certain exceptional circumstances, for an inverted bucket trap to lose its water seal. If, for example, it is used for draining a steam pipe carrying highly superheated steam, it is possible, once the system is up to temperature, for the water in the trap to be evaporated, when the trap will blow steam full bore and cannot reseal itself and will have no desire to close. When inverted bucket traps are used in such circumstances it may be desirable to connect them to the plant by a length of unlagged pipe. It also occasionally happens that the water seal is broken by reflux action. This is particularly likely if the trap has to be fitted above the plant it is draining. In most circumstances a non-return valve should be fitted between the trap and the plant it is draining.

Although in theory an inverted bucket trap should always work intermittently except when on full load, in practice a continuous discharge often takes place. There is no harm in this at low pressures, but at high pressures it is undesirable as valve or seat may be scored. The continuous action may be due to the trap or valve parts being incorrectly sized for the pressure and load under which it is working.

An inverted bucket trap must not be completely lagged. The top should be left bare in order to condense the steam that must be leaked off the bucket. The sides and bottom can be lagged.

Before an inverted bucket trap is set to work it should be filled with water to provide a seal.

285. THERMOSTATIC OR EXPANSION TRAPS. These traps distinguish between water and steam by the fact that water can be cooler than steam under the same pressure. It follows that they can never discharge condensate immediately it is formed. From this it follows that unless they are fitted to plant where the heated material can absorb some of the sensible heat, heat must be deliberately got rid of in order to cool the condensate to a point where the trap can make its choice.

They have, however, several great advantages. They are very small and light. They remove air with great readiness. They can, with one exception, withstand water hammer without injury. They are open and empty when the plant is shut down and are not therefore liable to damage by frost. They can work at any pressure however high because there is no inherent limitation as in the case of the buoyancy traps. They can work in conditions of movement and vibration that may be unsuitable for the mechanical traps. They are unsuitable for handling very large quantities of condensate, and they cannot discharge condensate at steam temperature.

286. METALLIC EXPANSION TRAPS. Fig. 125 shows a design of metallic expansion trap. Condensate enters the tube A and flows out past the valve C. The tube A is made of copper which expands considerably more than the iron body B. When the tube A expands it closes the opening at C by moving towards C. The valve C is pressed towards the tube A by the Spring E. The

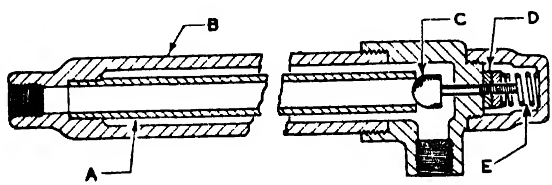


FIG. 125. METALLIC EXPANSION TRAP

two nuts D allow adjustment to be made so that the valve shuts tight at any desired temperature. The spring E allows the valve to be pressed back by undue expansion of tube A should the trap be fitted to a line where the temperature is higher than that for which the valve is set, or in case superheated steam reaches the trap.

The trap is very robust and simple, but, unless it is very large, its rate of discharge is small.

This trap is a pure thermostat. It opens at a certain (adjustable) temperature which must be quite definitely below steam temperature. The discharge is

irregular—it is neither truly continuous or properly intermittent. It is wide open at the start up and will therefore clear the system of air rapidly and effectively.

287. LIQUID EXPANSION TRAPS. Fig. 126 shows a liquid expansion trap. Liquids expand more than solids for a given rise of temperature, so that, by using a liquid, it is possible to get quite a fair valve movement by direct expansion. A liquid expansion trap can therefore be smaller than a metallic expansion trap. The tube A is filled with a suitable liquid—probably oil of some kind. One end of tube A is brazed into the cap B which carries a screw by which adjustment is made. The other end of the tube is brazed into another cap C which is brazed to the corrugated bellows tube D. The other end of the bellows tube is brazed to the piston E. The bellows tube D acts as a leak-proof gland between the liquid tube A and the piston rod F. The end of the piston rod is screwed into the valve G. When the liquid gets heated its expansion pushes out the piston and closes the valve.

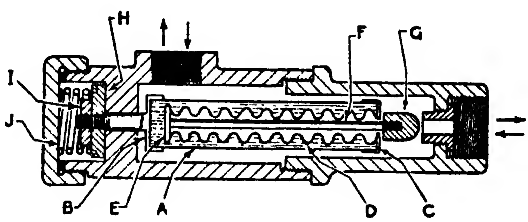


FIG. 126. LIQUID EXPANSION TRAP

The temperature at which the trap closes can be adjusted by screwing the nut H, after loosening the lock-nut I. This alters the length of the expansion unit and so adjusts the amount by which the piston must move to close the valve. The adjustment nut H is pressed tight against the body of the trap by the heavy spring J. If the trap is overheated, continued expansion after the valve has closed will push the expansion unit back against the spring J and thus protect the trap from damage. Safety precautions of this sort are necessary in both liquid and metallic expansion traps. The expansion force is to all intents and purposes irresistible. If the trap were adjusted to close at saturation temperature and by some chance superheated steam were to reach the trap there would be nowhere for the expansion to go were there no safety spring. Even if superheated steam did not reach the trap there may well be conditions where extra expansion takes place. It takes some little while for the element to get up to temperature. If the trap is not to pass steam while acquiring extra heat, it must be set to close before it has been completely heated up. After the valve has closed the element will still go on absorbing heat and will continue to expand.

This trap is a pure thermostat. It opens and closes at a certain (adjustable) temperature. The discharge temperature must be well below steam temperature. It is wide open when cold, and discharges air freely and efficiently.

It is robust and will withstand water hammer. It can be made suitable for any pressure. It is somewhat sluggish in response and is not made in sizes to handle very large quantities.

Traps of the metallic or liquid expansion types can be fitted so that the expansion element is either on the outlet or the inlet side. If it is fitted with the element on the outlet side a little steam will continually be allowed to pass. This steam will keep the element warm and close the valve. If the element is downstream of the valve the steam pressure is available to help to open the valve ; this may sometimes be advantageous.

288. BALANCED PRESSURE EXPANSION TRAPS. Although these traps operate by thermal expansion, they work on a totally different principle to the last two types. Fig. 127 shows a balanced pressure expansion trap. The expansion element A consists of one or more capsules or bellows. These are filled with a liquid (for example, a mixture of water and alcohol) that boils at a temperature lower than water. When the element is heated, the liquid inside it boils, part of it vaporises and the element expands and so closes the

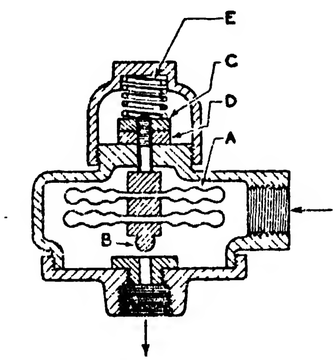


FIG. 127. BALANCED PRESSURE EXPANSION TRAP

valve B. Although the boiling distends the bellows due to the increase of pressure inside the element, this internal pressure is opposed by the pressure outside. Suppose the trap is used on much higher pressure condensate. The boiling liquid inside it will exert a much higher pressure, but this is opposed by the much higher pressure acting on the outside. There is thus a constant expansion movement at all pressures and the amount of this movement is fixed by the boiling point of the liquid inside the element compared to that of water. At very high pressures the pressure inside the element is very high, but, although the element is very fragile, this does not matter because the element is prevented from bursting by the very high pressure outside.

In the design shown in Fig. 127 nuts C and D are provided for adjustment, and spring E makes some provision for preventing damage to the element due to overheating at too low a pressure.

If superheated steam reaches the element the whole balance of forces is upset. The liquid inside boils vigorously and expands greatly with high internal pressure which is not balanced by a high external pressure. Balanced pressure traps must never be used where there is any possibility of superheated steam reaching them. The bellows or capsule is thin and fragile and cannot withstand corrosion, so these traps must not be used where the steam has any corrosive properties. The fragile element will not stand up to the shock of water hammer and must not be used where water hammer is a possibility.

The discharge from these traps is irregular. It is not inherently intermittent, but there is a lag while the liquid is taking in or giving up heat which causes the discharge to be semi-continuous.

The trap will open whenever it is in contact with anything that is cooler than condensate at the pressure at which the plant is working. It will therefore pass air whenever the air has cooled to below steam temperature. Balanced pressure traps are particularly suitable for use as automatic air vents as they are more responsive than liquid expansion traps. Balanced pressure traps are very light, small and cheap. They are not suitable for handling large quantities.

289. RELAY TRAPS. The foregoing consideration of the various kinds of direct-acting traps has shown that no trap has so far been described suitable for handling very large amounts of condensate, especially at high pressure. The float and bucket traps are limited by the valve area and weight or buoyancy of the float or bucket. The thermostatic traps will not handle condensate at steam temperature, although they have the inherent quality of an almost limitless force with which to work the valve. For passing large quantities of hot condensate at high pressures relay operated traps are almost essential. There are three types of relay trap :—

(*a*) The self-contained, steam-operated relay traps.
(*b*) The pilot trap.
(*c*) The servo-operated traps.

290. SELF-CONTAINED STEAM-OPERATED RELAY TRAPS. Fig. 128 shows a design of lever float relay trap of the self-contained type. When the float rises it opens the valve A and admits steam behind the piston B. The piston B being bigger in area than the water valve C will open the water valve and allow the condensate to be blown out. This design calls for a float of only sufficient buoyancy to open the very small valve A which can impulse the water valve C which can be of any size desired. There is, of course, an upper limit. At high pressure a more buoyant float will be needed to open valve A but valve A can be smaller at higher pressures, so that the limit is very high indeed.

Unless there is appreciable leakage past the piston this trap will give intermittent discharge. If there is moderate piston leakage the trap will work continuously, with a continuous waste of steam equal to the leakage. If the piston leak is serious the trap will not work.

If the piston leaks, the trap will discharge air. If the piston is tight it will not.

If the valve A is fitted below the water level, the trap becomes hydraulically operated, will waste no steam and should work continuously. Unless the piston leaks the valve C will not close after valve A has closed. To ensure satisfactory operation as an hydraulic trap a pin hole should be drilled in the piston.

Any of the mechanical traps, float, open bucket or inverted bucket will work as relay traps if suitably designed.

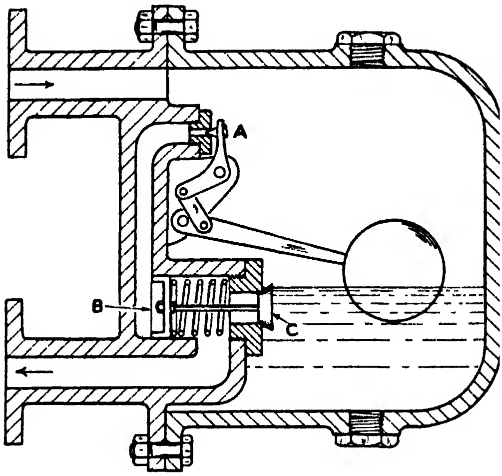


FIG. 128. SELF-CONTAINED STEAM OPERATED RELAY TRAP

291. PILOT TRAPS. If a small float trap A is connected up as shown in Fig. 129. it works as follows. The amount of condensate discharged by the small trap varies with the height of the float. The condensate thus discharged goes into the operating cylinder B of the main valve C. A leak-off D from cylinder B causes the main valve to tend to shut during low discharge periods. By making the leak off adjustable and connecting the adjustment to the main valve C it is possible to make the trap discharge quite continuously and evenly. The connection between valve C and the leak-adjusting valve E provides complete compensation. (Compensation is a necessary part of all automatic controls, and is discussed in Section 527.)

292. SERVO-IMPULSED TRAPS. Fig. 130 shows a large trap capable of handling 200,000 lb. of condensate per hour at 70 psi.g. pressure and discharging it into a flash tank at 15 psi.g. The vessel A is a reservoir or condensate receiver. At some suitable level, say half-way up, a small steel bottle B is connected to the vessel A by the two flexible pipes C. The bottle is suspended by the spring D, and actuates the hydraulic impulse valve E. Pressure is thus applied to the hydraulic cylinder F which opens or closes the condensate discharge valve G. When the hydraulic pressure is released the weight H closes

the valve G. Compensation is provided by the main valve G adjusting the tension of the spring D. Two traps of this design are in operation in the author's factory and handle almost the whole of the process condensate. (The method by which one large trap can satisfactorily handle condensate from a number of vessels working at different pressures is dealt with in Section 305.)

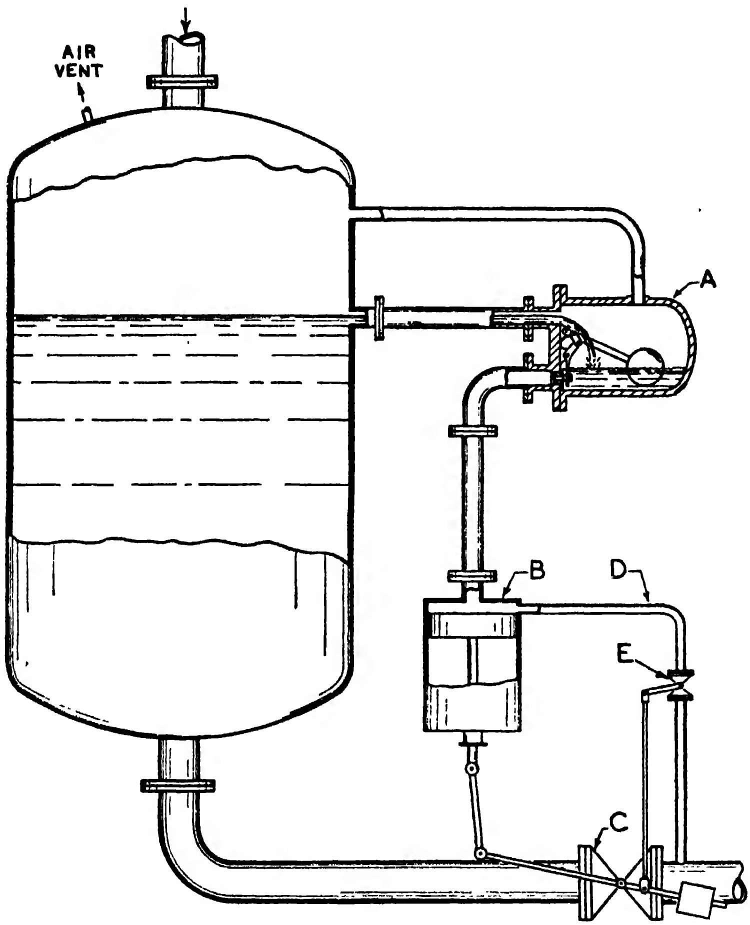


FIG. 129. PILOT TRAP

Another design of servo-trap is shown in Fig. 131. The trap is a float trap with a solid aluminium float A. The need for a gland is obviated by the use of the corrugated bellows tube B. The solid float is counterbalanced by spring D. The float lever operates the hydraulic impulse valve C which works the main condensate discharge valve. Four such traps, each capable of handling 50,000 lb. of condensate per hour at 250 psi, are in use in the author's factory. No compensation is provided; the trap body is much too small for the amount of throughput; the traps work intermittently and hunt badly.

293. PUMPING OR LIFTING TRAPS. Where condensate has to be lifted to a higher level, this can often be done by an ordinary trap, which uses the steam pressure, inside the heating surface that is being drained, to raise the condensate. There are many disadvantages attendant on this practice which, although very wide-spread, is generally to be deprecated. Some of these disadvantages have already been dealt with in connection with the drainage of steam mains in Section 190 and will be dealt with in connection with the U tube in Section 309. Condensate, in a well-arranged plant often reaches the hot well at such a high temperature that the boiler feed pumps cannot handle it. (See Table XLIV.)

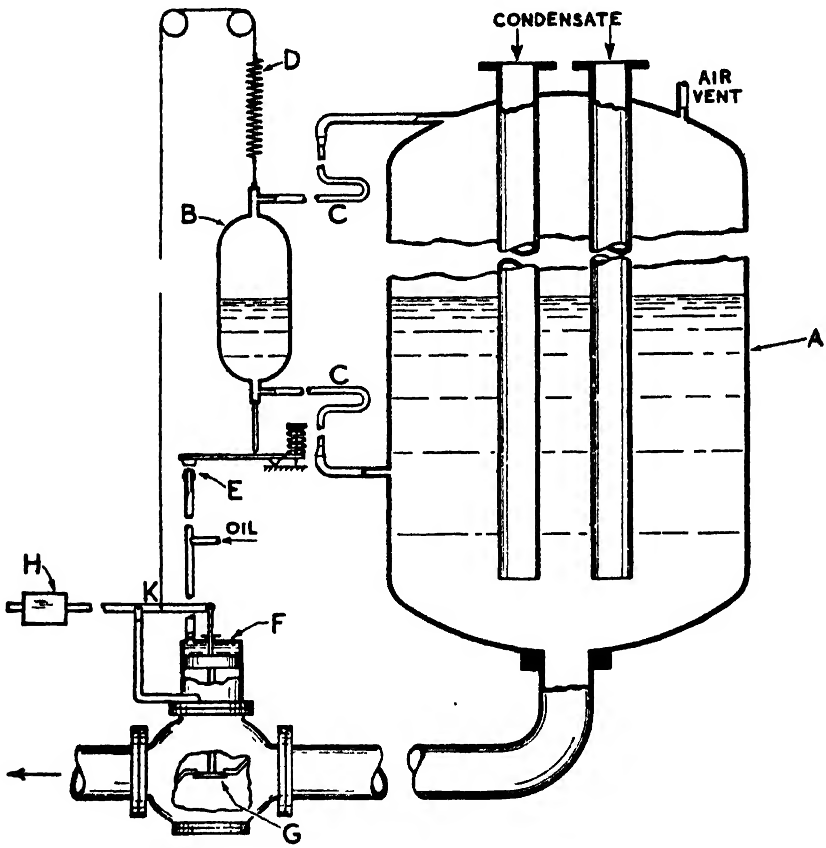


FIG. 130. LARGE TRAP WITH SUSPENDED BOTTLE RELAY

If the boiler feed pump supply tank is raised, the pump can then handle the hot condensate satisfactorily. Condensate at high temperature can be lifted very well by means of "pumping" or "lifting" traps. These machines are not really traps at all. They are pistonless pumps and are more akin to the "monte-jus" well known to the sugar industry.

The reason why a pumping trap can handle condensate that cannot be pumped mechanically is that there is no " suction " at the inlet of a pumping trap. The condensate, however hot, flows in by gravity. In any mechanical pump there is always some reduction of pressure at the pump suction. This causes flash which fills the pump with steam instead of water.

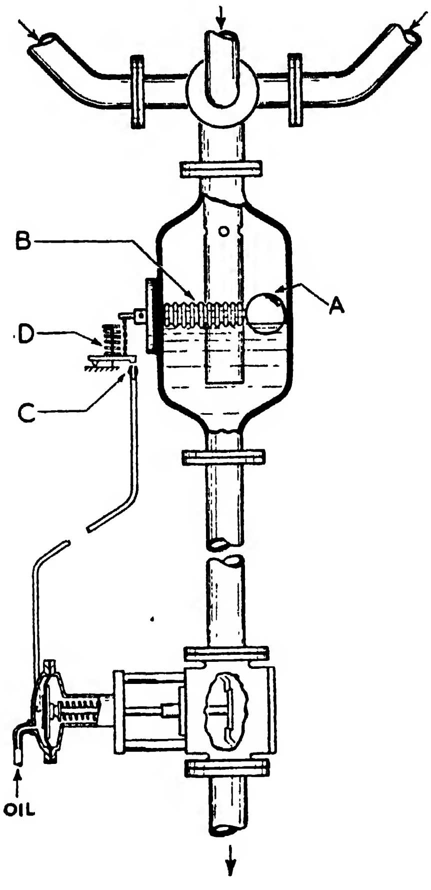


FIG. 131. LARGE TRAP WITH RELAY OPERATED BY SOLID ALUMINIUM FLOAT

Fig. 132 shows such a pumping trap. It is essentially a trip action float trap as regards its mechanism, similar to the trap shown in Fig. 121, but the mechanism trips two valves instead of one. The inlet and outlet pipes are fitted with non-return valves. The trip mechanism instead of opening a water discharge valve trips a steam inlet and a steam exhaust valve. In the position shown in Fig. 132 the condensate runs into the trap by gravity but cannot get out because the outlet non-return valve is held closed by the back-pressure in

the condensate rising pipe. When the float rises sufficiently to trip the mechanism, the steam exhaust valve shuts and the steam inlet valve opens. Steam pressure (any suitable pressure) closes the inlet non-return valve, opens the discharge non-return valve and blows the condensate through the outlet. When the condensate has been discharged the float drops, trips the mechanism and the steam is blown off through the steam exhaust valve. As soon as the trap has blown off its pressure the condensate can again flow into the trap.

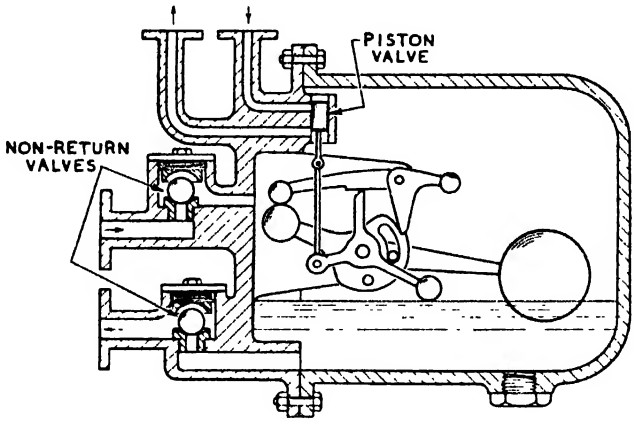


FIG. 132. PUMPING TRAP

The steam consumption of such a trap must be at least equal to the condensate in volume, plus any steam condensed by radiation or condensed in the condensate. The exhaust steam from such a trap should be piped through a coil in the high or low condensate tanks or can be led to a spray condenser or used in a very low pressure main. It is probable that the real steam consumption of such a trap is about four times the volume of the condensate handled. Provided the exhaust steam is used, the efficiency is high ; the only loss is that due to radiation. If, however, the exhaust is blown to atmosphere, the efficiency cannot be more than that of a direct acting pump and may be much worse. See Section 116 and Fig. 36. If the trap is pumping against a head of 45 ft. the pressure inside the trap will probably be about 25 psi.g. At this pressure the volume of steam is about 600 times that of water. So we can expect that 1 lb. of steam will lift between 150 and 300 lb. of condensate 45 ft.

A trap of this kind can be used to remove condensate from a heating surface working under vacuum where there is insufficient headroom to instal a barometric leg and atmospheric tank—see Section 308. In order that the condensate may run in freely, the pressure inside the trap during the filling part of the cycle must be the same as that in the heating surface. This is achieved by connecting the exhaust of the trap to the heating surface. This is bad from an entropy-increase point of view, but is the only practical solution so far as the author is aware. The arrangement is shown in Fig. 133. The pumping trap and the

barometric leg, Section 308, are not the only two choices for the removal of vacuum condensate. The Edwards pump and special centrifugal pumps can be and are being used.

294. SELECTION OF TRAPS. It is hoped that the descriptions of the various kinds of traps, their peculiar properties, good qualities and limitations will have dispelled the idea that a trap is just a trap. Every different kind of trap has certain qualities that make it particularly suitable for application to a particular type of installation. The choice of steam trap deserves just as much care as does the choosing of a good dinner or a gift to one's beloved. Table XLIII gives a summary of the characteristics of the various kinds of traps with some indication of their limitations and applications.

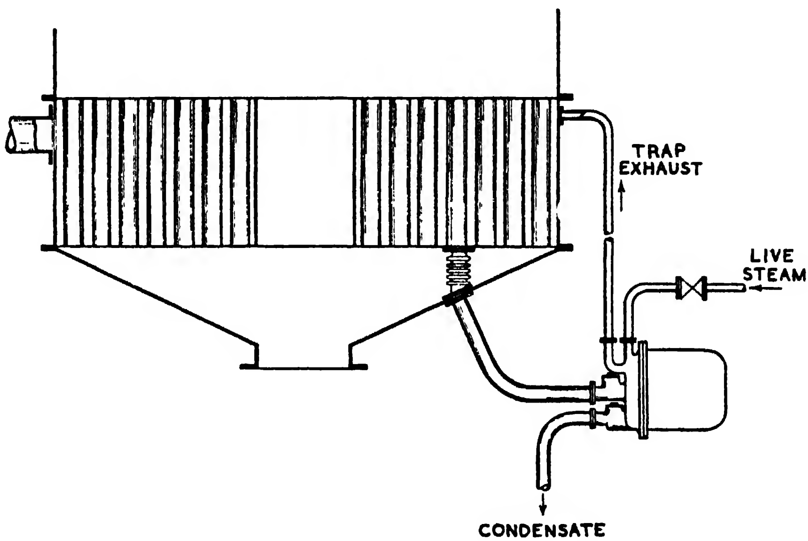


FIG. 133. PUMPING TRAP DRAINING VACUUM SPACE

Some guidance can be given. Wherever condensate must be removed instantly at steam temperature a mechanical trap must be used. Unless continuous discharge is desirable a bucket trap may be better than a float trap as it has no float to puncture and waterlog. Unless great quantities of air must be vented quickly the inverted bucket trap will get rid of moderate quantities of air quite quickly enough for most applications and thus eliminates the need for separate air vents. When the plant to be drained gives an extremely small condensate flow there is probably no mechanical trap small enough to be properly loaded. In such cases an expansion trap should be used. If a thermostatic trap is used where plant must be drained instantly the trap should be connected to the plant by a length of unlagged pipe. When sensible heat can be usefully taken from the condensate, as in any oversize air heater, a thermostatic trap is essential. Where great quantities of air must be quickly removed from a small plant, a thermostatic trap is almost a sine qua non.

TABLE XLIII. CHARACTERISTICS OF STEAM TRAPS

TYPE OF TRAP	TYPE OF DISCHARGE	OPENING FORCE	CLOSING FORCE	TEMPERATURE OF CONDENSATE	DISCHARGE AIR	WITHSTAND WATER HAMMER	STRAINER BEFORE TRAP	CONDENSATE DRAINED	WILL LIFT CONDENSATE	DAMAGE BY FROST	CHECK VALVE BEFORE TRAP	SUITABLE FOR SUPERHEATED STEAM	SUITABLE FOR VARYING PRESSURE
Plain float.. ..	Continuous	Buoyancy	Float weight	Saturation	No	No	Highly desirable	Instantly	Yes	Yes	No	Yes	Yes
Trip float	Intermittent	Buoyancy	Float weight	Saturation	No	No	Desirable	As formed	Yes	Yes	No	Yes	Yes
Open bucket ..	Intermittent	Weight of bucket	Buoyancy	Saturation	No	Yes	Not essential	As formed	Yes	Yes	Yes	Yes†	Yes
Inverted bucket ..	Intermittent	Weight of bucket	Buoyancy	Saturation	Yes	Yes	Not essential	As formed	Yes	Yes	Yes	Yes†	Yes
Metallic expansion	Semi-continuous	Metallic contraction	Metallic expansion	Pre-set temperature	Yes	Yes	Highly desirable	At pre-set temperature	Yes	No	No	Yes	No
Liquid expansion..	Semi-continuous	Steam pressure	Liquid expansion	Pre-set temperature	Yes	Yes	Highly desirable	At pre-set temperature	Yes	No	No	Yes	No
Balanced pressure expansion.	Semi-continuous	Differential pressure	Differential pressure	Below saturation	Yes	No	Highly desirable	After cooling	Yes	No	No	No	Yes
Relay—float, bucket, bottle.	Continuous if compensated	Outside source unlimited	Outside source unlimited	Saturation	No	—	Desirable	As formed	Yes	Yes	No	Yes	Yes
Pumping or lifting	Intermittent	—	—	Any temperature below steam temperature	No	—	Not essential	As formed	Yes	Yes	In trap	Yes	Yes
Barometric leg* ..	Continuous	Always open		212° F. or less	No	—	Unnecessary	Instantly	No	—	No	—	Yes
U-tube*	Continuous	Always open		Saturation at exit pressure	No	—	Unnecessary	Instantly	No	—	No	—	No

* These are not strictly traps; they are piping systems which replace traps—see Chapter 9.
† The water sealing the trap may, in exceptional cases, be evaporated—see Section 284.

All types of trap have not been described. There are, for example, the labyrinth traps and those float traps where the float is not under pressure. The former are simply slight improvements on an open but constricted discharge. The latter cannot be used for discharging condensate above their own level, and as they have their valve on the plant side of the float the float cannot distinguish between water and steam until the valve has opened ; consequently such traps will waste a little steam.

295. AIR LOCKING AND STEAM LOCKING. If a trap fills with air which cannot escape, clearly it cannot act properly. There are only two types of trap that have the inherent property of removing air. The expansion traps will remove an unlimited amount of air. The inverted bucket traps will remove moderate amounts of air. The removal of air from heating surfaces, condensate systems and traps is dealt with in Chapter 10.

No trap, however good and willing, can remove condensate if the condensate cannot freely reach it. There is a condition, which is pretty frequently met, where steam can act as plug between the plant to be drained and the trap, and, until this steam is condensed, no condensate can reach the trap. It is also possible, where more than one vessel is connected to one trap, for unbalanced pressures in the condensate pipes to prevent the condensate from one of the vessels reaching the trap. In all these cases, only too often, the trap is blamed, whereas it is really innocent ; it is the lay-out that is wrong. Steam locking and group trapping are dealt with in the next Chapter when discussing condensate handling.

296. FROST. Traps are often fitted outside buildings in alleys and sumps and other horrible places. During periods of shut-down the trap may freeze and get damaged. When the plant is shut down the steam condenses and the pressure drops. All the condensate runs into the trap but there is no steam pressure to discharge it, so that in a state of shut-down mechanical traps are generally full of water. Northcroft says that thousands of traps are destroyed every time this country has a long spell of frost. Thermostatic traps, for the most part, are self draining and do not suffer damage from frost as their valves are always open when they are cool.

The cure is to put all traps in a proper place inside a building as close as possible to the plant they are draining, and lag them carefully. Traps in " Steamy Alley " are generally wasting condensate and flash steam and seldom get proper maintenance.

297. STRAINERS. Float traps and thermostatic traps should always be preceded by a strainer. One of the fruitful causes of trap failure is scale, tow, dirt, sand, mice or other souvenirs blocking the valve or causing the valve to stick open or leak. Two forms of strainer are shown in Fig. 134. In Fig. 134a there is no dirt pocket, or rather the pocket is rudimentary ; but the dirt collects on the outside of the gauze. In Fig. 134b there is a good dirt pocket but the dirt collects on the inside of the gauze. If most of the foreign material is fibrous it is easier to remove from the outside of the gauze, but if it is sand or small scale it is more easily washed off the inside. Strainers *must* receive

regular attention, else they will choke, and again the trap will be blamed and will probably be bye-passed. Top discharge traps, i.e. the bucket traps, may often be better without a strainer. The strainer is an additional cost and an extra maintenance requirement, and unless the foreign material can float, the scale, etc., will remain in the bottom of the trap.

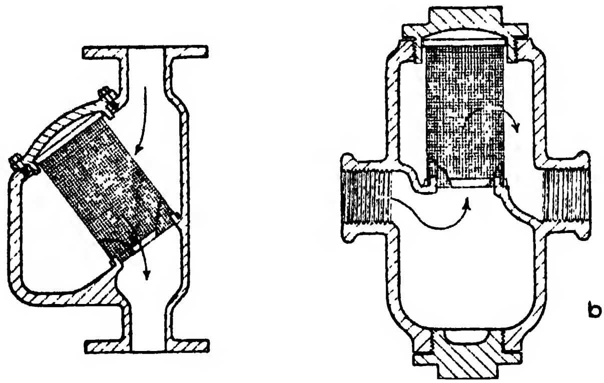


FIG. 134. TRAP STRAINERS

298. TRAP CAPACITY. Traps are often ordered by the size of their pipe connections. Pipe connections are like the flowers that bloom in the spring—they have nothing whatever to do with the discharge capacity of the trap. The discharge capacity depends only on the effective area of the valve and the pressure drop across it, together with the temperature of the condensate. When hot condensate is discharged from a trap, flash steam starts coming off immediately the pressure is reduced, that is, inside the valve. Such flashing water occupies a greater volume than cool water and limits the output of a particular valve. The discharge rate can be worked out, but why bother ? Specify the discharge required and the minimum and maximum pressures. The onus is then on the trap manufacturer, who has all the necessary data at his fingers' ends.

* * *

A trap is just a special kind of automatic valve, which is capable of saving or losing a lot of money. It is well worthy of being chosen with care, installed with intelligence and looked after with zeal. Design, materials and workmanship are much more important than price.

In the next two chapters some of the applications of traps are discussed. In most cases the plain float trap has been used for illustrative purposes. This does not mean that the plain float trap is necessarily the best. It was chosen for illustration because its action is the simplest of all traps, because it works instantly, and because it does not vent air.

* * *

CHAPTER 9

CONDENSATE HANDLING

> My doctrine shall drop as the rain,
> My speech shall distil as the dew,
> As the small rain upon the tender herb.
>
> DEUTERONOMY. XXXII 2. B.C. 700

CONDENSATE is formed when steam condenses, either accidentally—inside a steam pipe—or purposely—inside a heating surface. In most cases condensate must be removed as rapidly and completely as possible. The danger of allowing condensate to accumulate in a steam pipe has been discussed in Chapter 5, Sections 190 and 191. Unless condensate is quickly and effectively drained from heating surfaces the steam cannot make full use of the surface and output suffers. There may be occasions when partial waterlogging of a heating surface is permissible, first to enable some of the sensible heat to be removed from the condensate, secondly for purposes of control. These conditions are dealt with later.

299. RESISTANCE OF WATER FILM TO HEAT TRANSFER. In Chapter 4, Section 162, Table XIX, the heat conductivities of various substances are given. It will be seen that water is a very bad conductor indeed. A film of water 1/100 in. thick offers the same resistance to heat transfer as does a copper plate 6½ in. thick, or a steel plate ¾ in. thick. It is important therefore that heating surfaces should be drained as rapidly and completely as possible and that they should be arranged so that the condensate film is as thin as possible.

300. ARRANGEMENT OF HEATING SURFACE FOR DRAINAGE. There are three forces acting on the condensate that forms on the heating surfaces of a piece of plant. The first is gravity, the second is the frictional drag or brushing action of the steam flow ; the third is surface tension which is discussed later—see Section 348. These indicate two general principles. The steam should always flow in the same direction as the condensate ; and we can say that heating tubes or jackets should if possible be vertical or steeply sloping so that the steam inlet should be at the top. In the majority of plant this can be arranged, for example air heaters, unit heaters, calorifiers, boiling pans, evaporators, stills, etc. In many commercial designs of plant the heating surfaces are horizontal. This is done because it is often easier to get good heat transfer to the material being heated if the convection currents in the heated material are at right angles to the heating surface, and this advantage may outweigh the advantages of good condensate draining. The sag in horizontal tubes is often beneficial in absorbing expansion and contraction.

Some types of plant cannot be arranged with vertical or steep heating surfaces nor with concurrent steam and condensate flow. (One of the most successful of these—it breaks both rules—is the Lillie film evaporator.) The principal representatives of this class are the various drying cylinders, calenders, bowls, cans, ironers, decoudens, paper machines, food driers, etc. Another type of horizontal heating surface is found in the platens of moulding presses. In these platens the steam and condensate flow in the same direction but the

flow is so sluggish that it exerts no appreciable effect on the horizontal condensate film. There are at least two solutions to the press platen problem ; the drying cylinder is far more difficult.

301. DRAINING HORIZONTAL SURFACES. There are some horizontal heating surfaces where drainage is easy, for example, the bed of a laundry calender or multiroll ironer, the lower half of an ironing press or the bottom platen of a moulding press. In these plants only the top surface is true heating surface, so that, provided there is ample room inside, half an inch of water does no harm. Multiplaten presses are much more difficult. Each platen has two working surfaces. Unless the condensate can be very well drained there is bound to be a temperature difference between the upper and lower surfaces. This means that the top of each mould will be cooler than the bottom. It is almost impossible to give a fall, inside a platen, to enable the condensate to run out easily. This would mean thicker metal, with consequent lower heat transfer, on one side than the other. It would also call for thicker platens with resultant cutting down in press capacity. The multiplaten press is sometimes improved by steam circulation, but an even better solution may be the use of pressure hot water (see Chapter 18).

Horizontal coils and banks of tube should not be horizontal ! They should have as large a fall as can be permitted and accommodated.

302. DRAINING DRYING CYLINDERS. The draining of condensate from drying cylinders is the most difficult of all condensate handling problems. Fig. 135a shows the horizontal cross-section of a typical drying cylinder. The right-hand trunnion is fitted with a special gland. Steam enters through the outer ring of the gland. The inner part carries the condensate discharge pipe which dips into the bottom of the cylinder. In Fig. 135a the discharge pipe is shown dipping into a pool of condensate which is being blown out by the steam pressure into the trap and away. When the level of the condensate falls sufficiently steam passes up the dip pipe into the trap and the trap closes. This condition is shown in Fig. 135b where the dip pipe, trunnion pipe and trap are all full of steam. Condensation continues and the condensate forms an increasing pool in the cylinder, but the dip pipe is full of steam so that the condensate cannot get out, as shown in Fig. 135c. Now once steam has got into the dip pipe or trap *it is already wasted*, it has passed through the cylinder without condensing. It must be got rid of before any more condensate can leave the cylinder. This can be done in two ways. Either by leaving the trunnion pipe and trap unlagged so as to condense the locking steam, or by deliberately leaking it off to waste.

In addition to live steam in the dip pipe there is some flash produced from the condensate. Suppose the cylinder in Fig. 135 is 5 ft. in diameter working with 5 psi steam. It takes over 1 psi to lift the condensate up the dip pipe. Consequently the pressure on the condensate in the trunnion will only be 4 psi. This will cause a small flash (about ·3 per cent.) which must be condensed or vented before condensate can be discharged.

Leaving it to condensation is haphazard and the rate of condensation will vary from day to day. The steam actually in the dip pipe cannot condense because it is in an atmosphere of live steam. All the condensation has to be done

by the trunnion pipe or the trap. Condensation of the locked steam will waste the minimum of steam. Leakage will be more wasteful but much more speedy, reliable and adjustable. As water is such a bad conductor of heat it is important that every possible assistance be given to the condensate to get away, so that a definite steam leak should be provided if a plain float trap is employed, or an inverted bucket trap may work well without any additional leak provision.

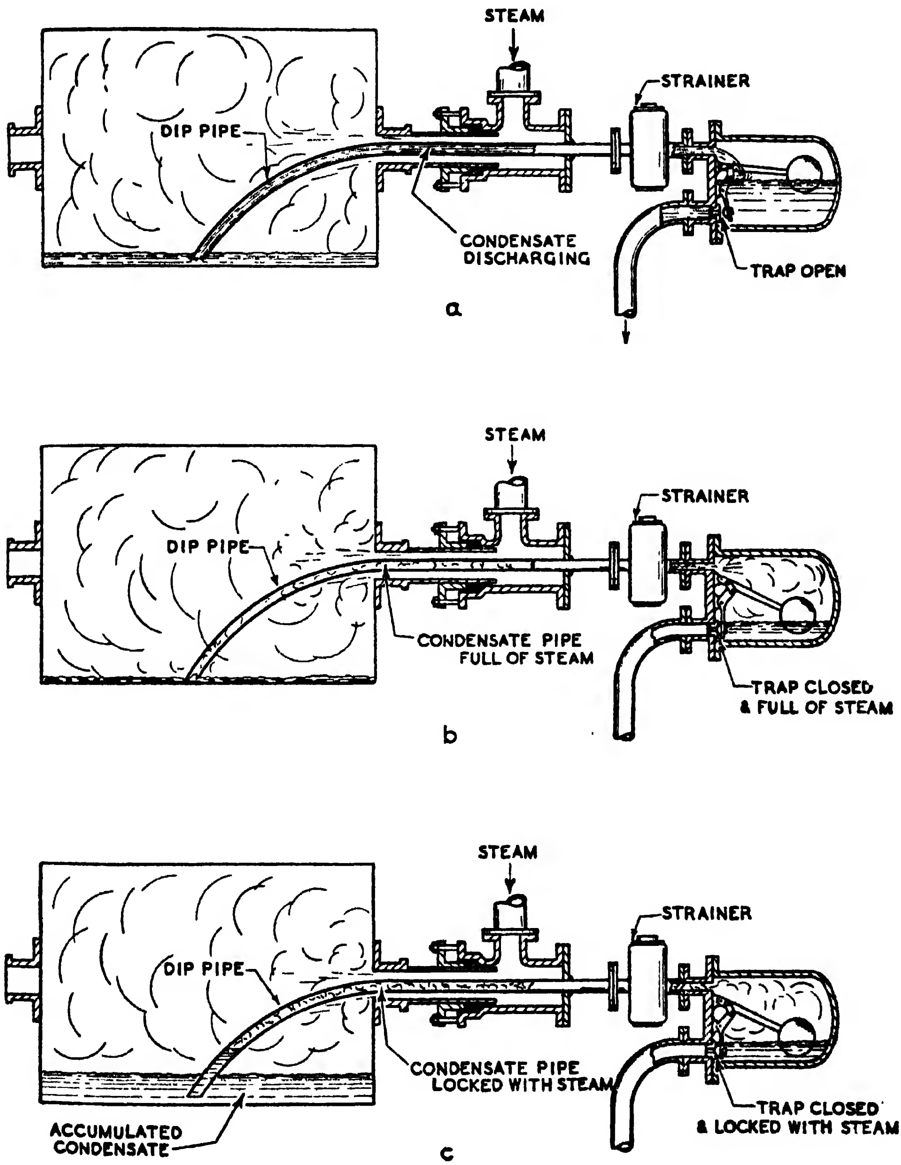


FIG. 135. DRAINING DRYING CYLINDER

The discharge from such a cylinder must obviously be intermittent, and the float trap is fitted, not for its continuous discharge properties but because it is the one that discharges condensate with the least delay.

The position of the dip pipe is important. Fig. 136 shows diagrammatically some cross-sections of drying cylinders. Fig. 136a shows the cylinder at rest with the pipe dipping into the condensate pool. Fig. 136b shows the effect

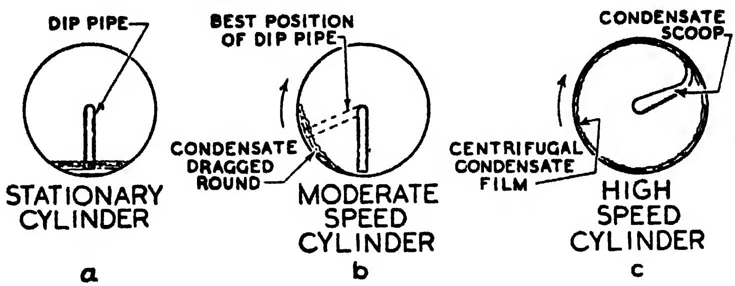


FIG. 136. CONDENSATE PIPE AND SCOOP IN DRYING CYLINDER

of running the cylinder. The condensate is dragged up clear of the dip pipe, which should be turned into the position shown dotted (this position can only be found by trial and error). This dragging up of condensate uses a lot of power. Northcroft quotes a paper mill which obtained a saving of 20 per cent. in power by reduction of the condensate in the cylinders of the paper machine.

In very fast running machines (this only applies to a few newsprint machines) the speed is such that centrifugal force maintains the condensate in an even film all round the inside of the cylinder. This film must be scooped off in some such way as is indicated in Fig. 136c.

Whatever the speed or size of the cylinder there will always be film of greater or lesser thickness of condensate on some or all parts of the inside of the cylinder. The smoother the inside surface the less power will be absorbed in dragging round the condensate and the nearer can the dip pipe or scoop be set to the surface, thus reducing the amount of condensate.

303. STEAM RELEASE FLOAT TRAP. In the previous section it was pointed out that any steam locked in the dip pipe and trap of a cylinder draining system must be wasted. A float trap fitted with a steam release is shown in Fig. 137. A small vent is provided above the normal water level with a needle valve adjustment. Like the inverted bucket trap this will always waste steam. In the inverted bucket trap this waste is limited, except during discharge, by the amount that the trap can condense ; whereas with the steam release the waste is the amount that the needle valve allows to escape. There is a compensation. In the inverted bucket trap the waste steam is condensed by radiation and all its latent heat is lost. With the steam release the steam passes into the condensate pipe and can be recovered and used if there is a flash collection system. Such a steam release will of course also remove air.

304. STEAM LOCKING. The condition described in connection with drying cylinders, where steam in the condensate pipe prevents the escape of condensate, is called steam locking. There are a number of other pieces of plant which are inherently steam-locking. In many factories there are glass-lined tanks or other special vessels through the bottom or sides of which it is undesirable that connections should be taken.

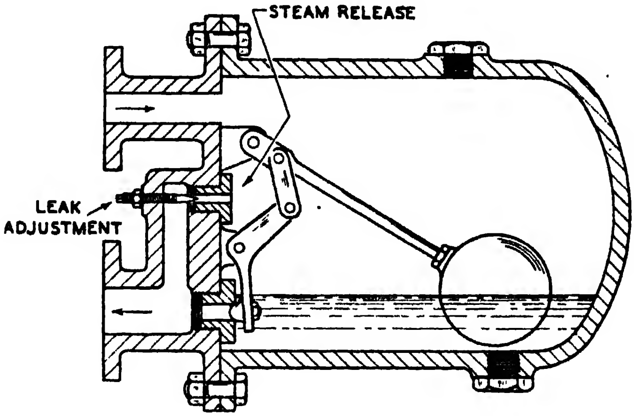


FIG. 137. STEAM RELEASE ON PLAIN FLOAT TRAP

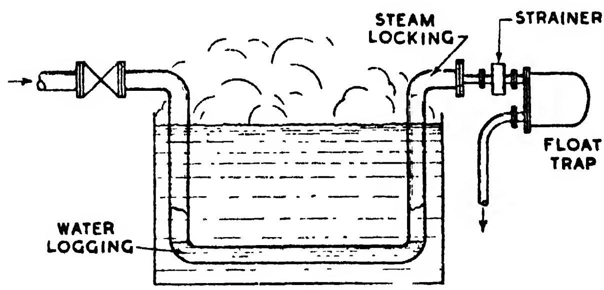


FIG. 138a. STEAM LOCKED COIL

A steam coil fitted in such a vessel is fed from the top and the condensate is discharged over the top in the manner shown in Fig. 138a. Condensate cannot be discharged until it has accumulated to such an extent as to fill the horizontal part of the coil. Before this accumulation can get to the trap the steam in the riser and in the trap must be wasted by condensation or leak. After each massive discharge all the heating surface will be effective and heat transfer will be brisk. As the coil waterlogs heat transfer will drop until, just before the next discharge, it becomes almost zero. Apart from slowing down the process such intermittent heating may be technically undesirable.

The cure is shown in Fig. 138b. A lever float trap with a steam release should be used. The coil is given a definite fall and a tee is fitted to the blind end. The bottom of the tee forms a condensate well dipping into which is a small bore riser leading to the trap. Waterlogging is prevented. The only steam that must be wasted is that in the trap and in the small bore riser. Such waste is only occasional as the condensate will be more nearly continuous in discharge. The tee is called a lift fitting and is shown enlarged in Fig. 139a. Most trap makers sell lift fittings, see Fig. 139b. Some of these fittings are not too satisfactory as the riser is relatively too large.

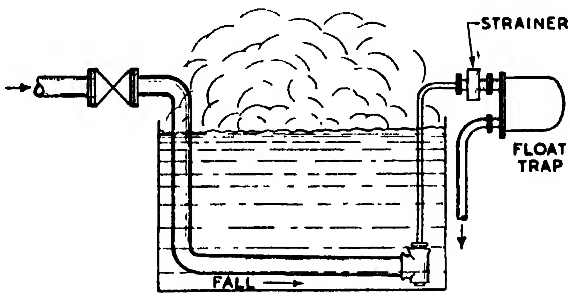


FIG. 138b. THE CURE FOR A STEAM LOCKED COIL

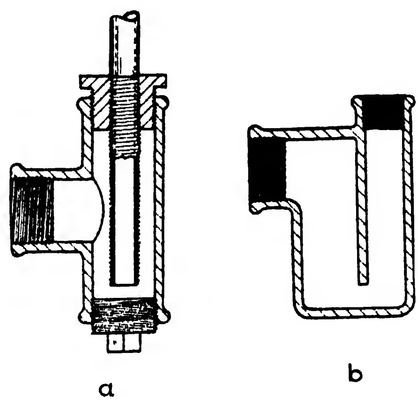


FIG. 139. CONDENSATE LIFT FITTINGS

Another common piece of plant that is inherently steam locking is the ordinary tilting cooking pan—jam, confectionery, soup, etc. The steam is admitted to the jacket through one trunnion and the condensate driven out through the other. Steam and condensate are frequently led to the bottom of the jacket by the cored passages shown in Fig. 140a. This design admits the steam in the wrong place, is inherently steam locking and consequently must waterlog and work intermittently.

Fig. 140b shows a different design. The steam is admitted evenly round the top so that it helps to sweep the condensate down. There is a good condensate well provided with a small bore outlet riser leading to a steam release float trap.

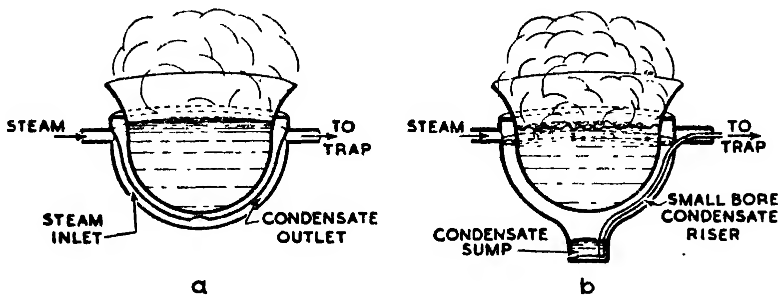


FIG. 140. DRAINING HEMISPHERICAL JACKETED PAN

Fig. 141 shows another common cause of steam locking. The trap is too far from the plant. The condensate pipe is too long and has sagged, and the whole thing is elaborately lagged. The cure is to bring the mechanical trap as close as possible to the plant to be drained. If an expansion trap is to be used then the condensate pipe must be some feet long and should be unlagged.

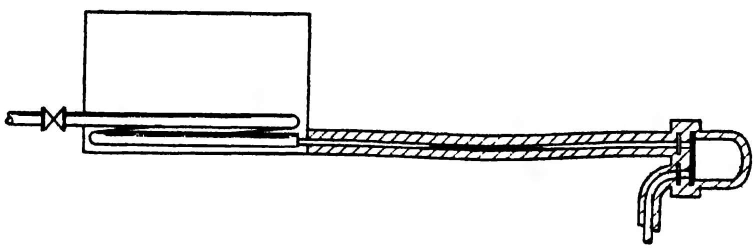


FIG. 141. WATER AND STEAM LOCK IN LONG SAGGING CONDENSATE PIPE

305. GROUP TRAPPING. It were far better, if possible, to use one large trap than a multitude of small ones. This great thought has occurred to many people who have put it into practice with, for the most part, unsatisfactory results. It is possible, in some cases, to replace a number of small traps by one large one ; in some cases it is undesirable. If one trap is to replace several it is necessary to comply with one vital requirement—the flow of condensate from any one vessel into the common trap must not be interfered with in any way whatever. Let us investigate what will happen if three identical coil-heated vats are connected, by a misguided enthusiast, to one trap.

Fig. 142 shows the arrangement. All three vats are doing the same job, say heating up a cold water solution to near boiling point and then holding the temperature for a while. Vat C has been on longest ; it has nearly reached

the appointed temperature ; heat transfer is low because the liquor is hot ; so that the steam pressure in the coil is nearly up to that in the main. Vat B has not been on so long ; the liquor is cooler ; condensation is more brisk ; consequently the pressure is a little lower than in coil C. Vat A has only just been put on ; the liquor is cold ; heat transfer is very brisk and condensation is heavy ; consequently the steam pressure in coil A is considerably below the main pressure. These pressures are the " natural " pressures. The pressure in the condensate main must be equal to the highest pressure connected to it, namely 19 psi. Coil B is at 18 psi and is 2 ft. 6 in. above the condensate main. Condensate will stand 2 ft. 4 in. up the condensate branch, but the condensate will just be able to flow into the condensate main. The natural pressure in coil A is 15 psi. This + 2 ft. 6 in. of water head gives 16 psi. Therefore no condensate can possibly flow from the coil that is producing most condensate ! Coil A will waterlog until heat transfer has been so reduced that the pressure in the coil rises to at least 18 psi.

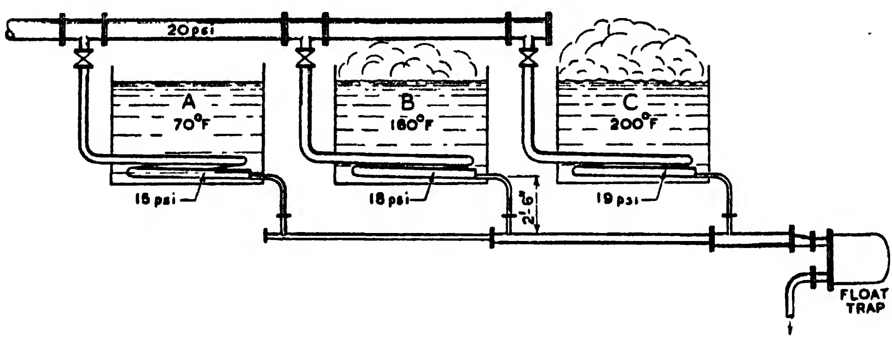


FIG. 142. GROUP TRAPPING—WRONG

The trap is blamed ; possibly a larger trap is fitted ; it still cannot work. Waterlogging cannot be blamed on the trap if the lay-out is such that the condensate cannot reach the trap. There are two solutions to the problem. The first easy but not satisfactory ; the second entirely satisfactory but of limited application.

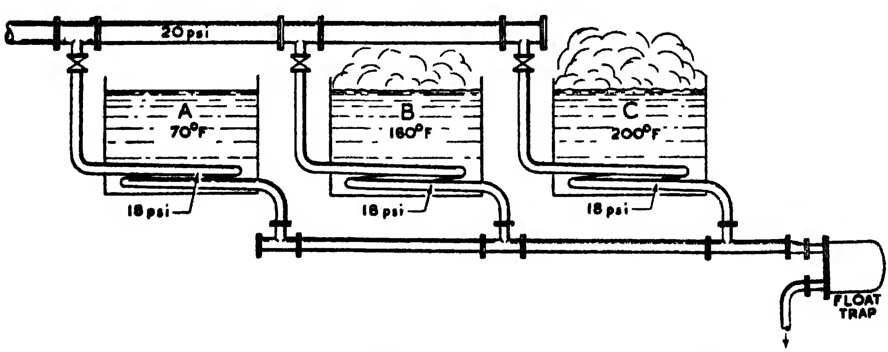


FIG. 143. GROUP TRAPPING—SLIGHT IMPROVEMENT

The first solution is shown in Fig. 143. The condensate outlets have been made the full bore of the coils and the condensate main is made very large. The result is that the condensate can flow along the bottom of the branches and the main and there can be no waterlogging. There is one important and undesirable effect. The pressure in all the coils must be the same at all times ; it will equalise through the steam space in the large condensate pipe. This will slightly lower the pressure in coil C but this will be compensated by the greater steam velocity through the coil. Steam will pass back to coil A. This will increase the pressure in coil A but the condensate will have to flow against a steam flow. There is an important consequence from this evening out of pressure. It is impossible to throttle down one vat to say 3 psi to keep it hot after it is up to temperature. All vats must carry the same pressure so that one cannot be satisfactorily slowed down without the others. The large pipes are more expensive and will be expensive to lag and will lose more heat.

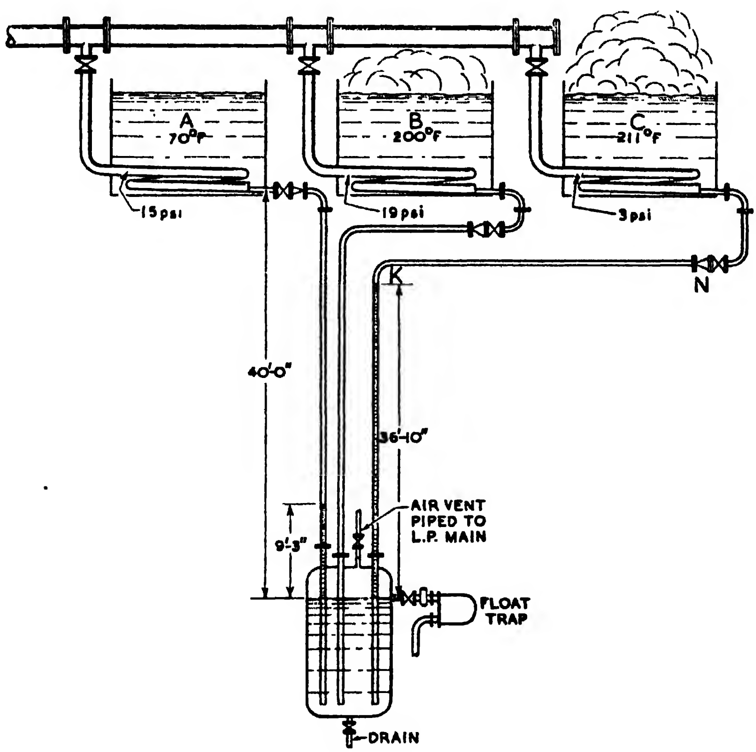


FIG. 144. THE ONLY CORRECT WAY OF GROUP TRAPPING

The right solution is shown in Fig. 144. It is free from all blemish but it requires height, more height than is available in most factories. The condensate pipes from each coil are led down and water-sealed into a common condensate tank at a sufficient distance below the vats to give a water head in the down pipes equal to the greatest pressure difference ever wanted between the vats. Suppose the maximum pressure on a coil is 20 psi and the minimum pressure

is 3 psi, the pressure difference is 17 psi, equivalent to 40 ft. of water—see Table XXXII. The condensate pipe from each coil is led into a trap tank 40 ft. below, and water sealed into the tank. Suppose Vat C is up to temperature and is to be kept warm with 3 psi of steam. Suppose vat B is nearly up to heat and has 19 psi in its coils, and suppose vat A is just starting and owing to heavy condensation has only 15 psi in its coil. All the coils can drain correctly and can maintain their own individual pressures. The condensate pipe from B will be empty except for the condensate running down it. The pressure in the condensate tank will be 19 psi. The condensate will stand 9 ft. 3 in. up the A condensate pipe, and it will stand 36 ft. 10 in. up the C condensate pipe.

A 15 psi + 9 ft. 3 in. = 19 psi.
C 3 psi + 36 ft. 10 in. = 19 psi.

The condensate or trap tank should have a capacity equal to the combined volume of all the condensate pipes. This prevents the water seal from breaking. The condensate pipes must run separately into the trap tank so that each can have its own individual water level to compensate for varying steam pressures. In order to retain the water seal the condensate pipes must reach nearly to the bottom of the tank.

This arrangement has been working satisfactorily in the author's factory for over ten years. The main process buildings are very high ; in one case the trap tank is 112 ft. below the condensate producers. This enables one vessel to operate at 60 psi and its neighbour to work equally satisfactorily at 12 psi while the condensate from both is handled by the same trap.

There is one snag that must be watched for. When one vessel is shut down, in order to prevent hot condensate being pushed back into the idle coil, non-return valves are fitted. If the steam stop valve on vessel C leaks, condensate will collect in a slug behind the non-return valve N. When the steam valve on C is opened, this slug may hammer the bend K. (This occurs occasionally in the author's factory.) The cure is better valve maintenance. The palliative is careful valve-opening. It is possible that the trouble might be cured by putting the non-return valve at the bottom of the pipe.

306. DRAINING UNIT HEATERS. Unit heaters have certain advantages. They represent the cheapest-first-cost method of space-heating large buildings. They take up very little room. They are suitable for high pressure steam (thermodynamically deplorable, but not necessarily so in this application). Consequently the steam piping can be very small and cheap. They are designed to be put in the head room, nicely out of the way. For tidiness sake the condensate main for collecting the water from unit heaters is very often fitted above the heaters. This is bad practice, but it is so often expedient that such upward draining must be considered. With any space heating arrangement, especially one working with high pressure steam, there is an important point that is sometimes lost sight of. The object of space-heating is to heat the space of the building. Therefore there is no need for elaborate lagging or any lagging of condensate piping where that piping passes through the building that is to be heated. This enables the sensible heat to be taken out of the condensate and possibly removes any objection to the use of high pressure steam, as matters can often be arranged so that there is no flash steam from the cooled condensate.

The condensate discharge from a unit heater is heavy and, while the heater is working, practically steady. Fig. 145 shows three of the many possible arrangements. The arrangement shown in Fig. 145a is the best, but if a high level condensate return is insisted on the lay-out shown in Figs. 145b and c will be satisfactory. There may be a risk of water hammer when discharging condensate upwards. This risk is much greater where flash steam formation is a possibility. Flash is always likely to occur at the trap discharge ; so that system 145b is preferable to system 145c. Provided a large enough expansion trap can be installed, a considerable amount of the sensible heat in the condensate will be used. The expansion trap will get rid of any air in the system, and an expansion trap is usually so small and light that it can be carried by the piping system without the need for separate supports.

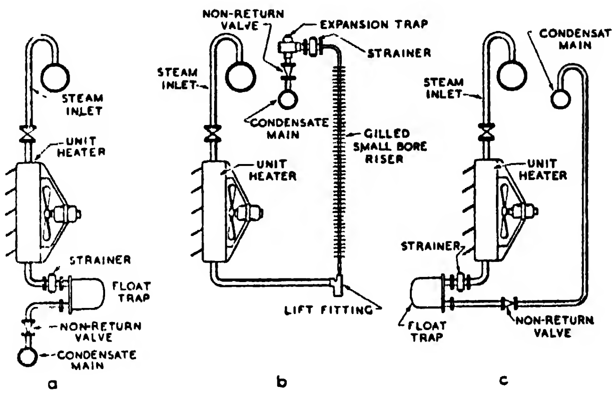


FIG. 145. DRAINING UNIT HEATERS

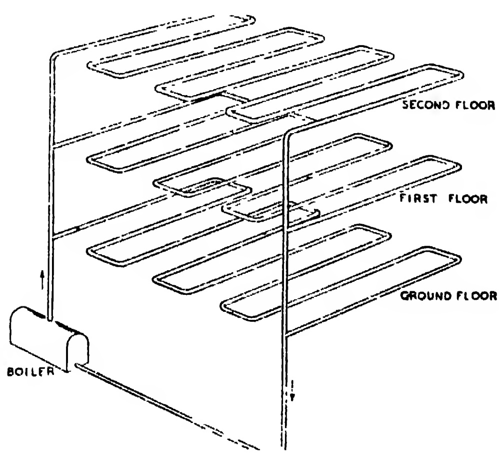


FIG. 146. GRAVITY RETURN SYSTEM HEATING THREE FLOORS

307. ONE-PIPE AND GRAVITY RETURN SYSTEMS. There are some extremely simple systems of condensate return that work, and have worked for a century. In some of these systems only one pipe is used, and the condensate returns by gravity without any trap or other device to the boiler. Fig. 146 shows a simple gravity system such as has worked for four or five generations in some textile mills. Were the fall on the pipes adequate, there could be little wrong with such a lay-out. In many of the older mills, however, there is not only insufficient fall but many dips and pockets. Sometimes each heating pipe is 1,000 or more feet long. With low buildings it is impossible to provide adequate fall on such a pipe. Water hammer is chronic and violent.

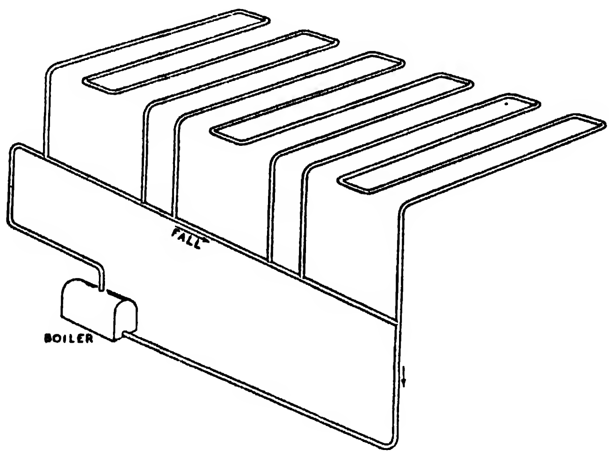


FIG. 147. ONE PIPE GRAVITY RETURN SYSTEM HEATING THREE ROOMS AT ONE LEVEL

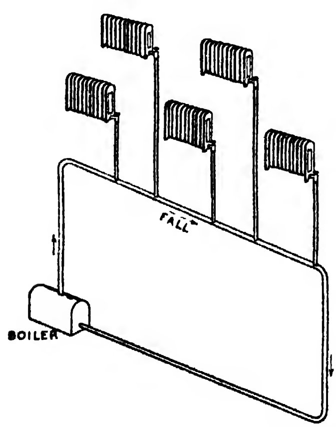


FIG. 148. ONE PIPE GRAVITY RETURN SYSTEM FEEDING RADIATORS AT TWO LEVELS

The system shown in Fig. 147 uses the steam main as the condensate return. Provided the main has an ample fall and is of large size there is no fault in such a system. Fig. 148 shows a one-pipe arrangement feeding radiators. This is not so good. It demands that each radiator be either full on or tight off. The radiator valves must be very large because the steam has to go in and the condensate has to get out against the steam flow. Such a system has worked but, to be satisfactory, pipes and valves have to be so large that there is little or no saving over a separate condensate system.

308. BAROMETRIC LEG AND ATMOSPHERIC TANK. Where the condensate is to be removed from a space working under a pressure below atmospheric pressure, the simplest of all devices can be used in place of a trap, provided the plant to be drained is sufficiently high above ground.

A vacuum of 20 in. means that the difference of pressure between the vessel under vacuum and the atmosphere will support a column of mercury 20 in. high. As shown in Table XXXII, 20 in. of mercury is equivalent to 22·7ft. of water. If therefore the condensate outlet can be led into an atmospheric tank in such a position that the barometric leg, or condensate pipe, is 23 ft. high, the condensate will run out freely by gravity into the atmospheric tank. See Fig. 149.

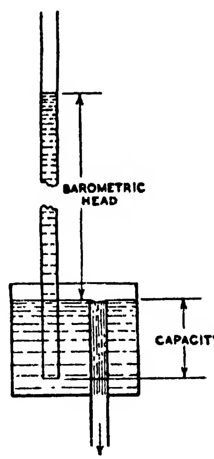


FIG. 149. BAROMETRIC LEG AND ATMOSPHERIC TANK

The barometric leg must extend nearly to the bottom of the tank, and the condensate must overflow from the top of the tank, so as to seal the barometric leg. The capacity of the tank between the ends of the two pipes must be greater than the cubic content of the barometric leg, to prevent the seal being broken when starting up.

309. THE U-TUBE. The U-tube has often been suggested, and has sometimes been used for removing condensate where the pressure difference between the inside and outside of the heating surface is very small. The results are by no means always satisfactory. The U-tube requires somewhat lengthy consideration.

In Fig. 150 a heating surface is supplied with steam at 10 psi.g. and the condensate is led to a U-tube with arms about 24 ft. long. The outlet end of the U-tube is connected to a condensate tank where the flash steam is to be condensed by a spray of cool make-up water. Now what happens ? Let us look at Fig. 151 which shows a diagrammatic U-tube with a very long outlet arm. Suppose the U-tube has 12 ft. of water in each arm at A in Fig. 151. Now suppose that the left-hand arm is put under a pressure of 10 psi.g. The level in the left-hand arm will drop and that in the right-hand arm will rise until the head between the

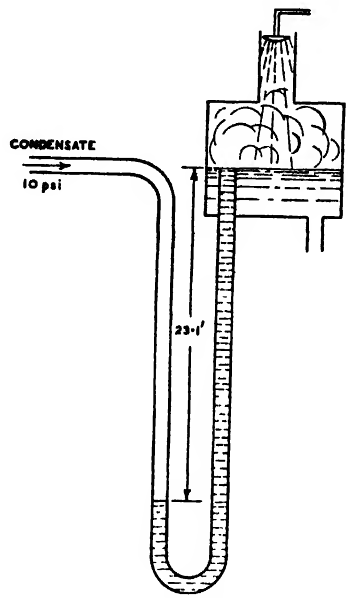


FIG. 150. U-TUBE WITH COLD SPRAY CONDENSING FLASH

two is 23·1 ft. at C in Fig. 151, and just balances the 10 psi pressure—see Table XXXII. If the water in the U-tube is condensate at 10 psi at saturation temperature, the water will be at 239° F. and have a heat content of 207·9 Btu/lb.

At point B_1 the water is at 10 psi and can exist at this temperature and heat content.

At a point 2·3 ft. above B_1, the pressure will only be 9 psi and water at this pressure can only contain 205·5 Btu.

So that 2·4 Btu must have gone in flash steam.

The latent heat at 9 psi.g. is 954·4 so that the amount of steam flashed will be $\frac{2{\cdot}4}{954{\cdot}4} = {\cdot}002514$ lb.

At 9 psi water has a volume of ·0169 cu. ft./lb.

Saturated steam has a volume of 17·2 cu. ft./lb.

So that the mixture of condensate and flash steam will have a volume of

$$(\cdot 002514 \times 17\cdot 2) + (\cdot 997486 \times \cdot 0169) = \cdot 0601 \text{ cu. ft./lb.}$$

instead of ·0169 were it all water.

Point B_2 will therefore be $\frac{\cdot 0601}{\cdot 0169} \times 2\cdot 3 = 8\cdot 4$ ft. above point B_1 instead of 2·3 ft.

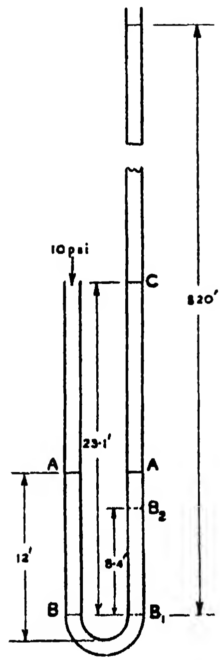


FIG. 151. THEORETICAL U-TUBE

Now the higher up the tube we go the more flash steam will have been liberated and the greater will be the volume occupied by the steam owing to its reduction in pressure. If we assume that the flash takes place evenly up the tube the average pressure will be 5 psi.g. where the volume of saturated steam and water are 20·4 and ·0168 cu. ft./lb. respectively.

The sensible heat in condensate at atmospheric pressure is 180·2 Btu, so that the heat to be got rid of by flash will be 207·9 − 180·2 = 27·7 Btu. The latent heat at atmospheric pressure is 970·6.

So that the weight of flash steam will be $\frac{27\cdot 7}{970\cdot 6} = \cdot 028539$ lb.

The volume of steam/condensate foam per lb. will be

$$\cdot 028539 \times 20\cdot 4 + \cdot 971461 \times \cdot 0168 = \cdot 5985.$$

So that the height of the foam column in the right-hand arm will be

$$\frac{23 \cdot 1 \times \cdot 5985}{\cdot 0168} = 823 \text{ ft.}$$

If our factory were situated beside the Eiffel Tower or the Empire State building we might decide to fit up a crazy U-tube 820 ft. high. If, however, our pressure dropped from 10 psi to 9·9 psi no condensate whatever would come out of the tube. If the pressure rose to 10·1 psi the column would be insufficient and the tube would be unstable and turn itself into a steam lift—just like an air lift well pump.

The calculation just given may be considerably wrong. 1 Btu of flash steam increases the height of the column by 30 ft. We are dealing with small differences in values given in the steam tables to four significant figures. The last digit in the steam table values is indicative rather than exact. Even if there is an error of 1 or 2 Btu it will not affect the point, which is that the hydrostatic column of flashing water will be a foam many feet high in a small bore pipe.

This somewhat laborious theorising is given in order to explain the difficulties that arise with a vertical pipe containing hot condensate. Apart from explaining the pitfalls that surround the use of a U-tube it explains the readiness with which water hammer will occur in any system where condensate is blown up to a higher level, and it shows that great care must be exercised in such systems to prevent the formation of pockets of steam driving slugs of water in front of them.

In actual practice a U-tube does not seem to behave quite as the foregoing theoretical argument would suggest. It works in erratic gushes. A bubble of flash steam forms, gets bigger by additional flash and by expansion, and throws out a slug of water.

It might be that were the rising limb of the U-tube made many times larger than the dropping limb, the flashing steam bubbles might separate and rise to the surface without greatly raising the liquid level. The author does not know whether this is true, though he has heard that the U-tube can give satisfaction in certain cases; they are used in many multiple effect evaporators for transferring the feed from body to body in forward feed—Section 481.

310. THE CONVECTION U-TUBE. Although the U-tube may not work in the form shown in Fig. 151 it can be and is being used in a modified form. Suppose we could mix with every part of 10 psi condensate 10 parts of condensate that have already cooled down to 212° F. by flash, by introducing this cooler water at the bottom of the U.

There will then be $\dfrac{(10 \times 180 \cdot 2) + (1 \times 207 \cdot 9)}{11} = 182 \cdot 7$ Btu/lb. in each of the 11 lb. of mixed condensate.

This corresponds to a saturation pressure of about 0·75 psi.g. or a water head of 1·7 ft.

We shall therefore have $\dfrac{182 \cdot 7 - 180 \cdot 2}{970 \cdot 6} = \cdot 002575$ lb. of flash steam at 25·6 cu. ft./lb. and 10·997425 lb. of water at ·0167 cu. ft./lb.—a combined volume of ·022684 cu. ft./lb.

The equivalent head of 1·7 ft. will be obtained from the steam/water mixture from a column $\frac{1 \cdot 7 \times \cdot 02268}{\cdot 0167} = 2 \cdot 3$ ft.

Our cooled U-tube will only need to be $23 \cdot 1 + 2 \cdot 3 = 25 \cdot 4$ ft., say 30 ft. high, instead of 800 ft.

A device, working on this principle, which is in operation in some beet sugar factories (and possibly other factories) is shown in Fig. 152. The hot condensate under pressure enters at the bottom of the wide bore pipe B.

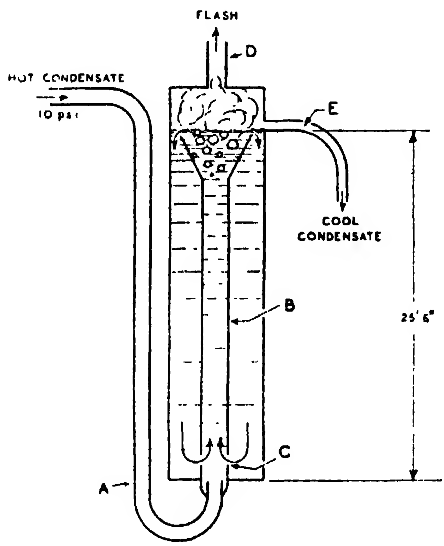


FIG. 152. CONVECTION U-TUBE

It flows upwards and flash takes place in the bell-mouthed portion of B, made this shape to prevent large steam bubbles from acting as steam lift pistons. The cooled condensate flows down the outside of tube B and enters the bottom of B through the holes C. This cools down the hot condensate and limits the flash zone to the top bell-mouthed portion. The flash steam is piped off through D and the condensate overflows by E. The success of the device depends on the maintenance of rapid convection currents to ensure heavy dilution of the incoming hot condensate. If this circulation gets upset the device becomes unstable and flashing takes place too low down and kicks slugs of water out of the top.

The design of a device such as this is more a matter of experimental art than science.

311. THE COOLED U-TUBE. Fig. 153a shows the downside leg of a U-tube cooled by means of a jacket through which water or some process liquor that needs heating is passed. This removes the excess sensible heat from the condensate and no flash has to be allowed for.

Fig. 153b shows another way of using a U tube on cooled condensate, by passing the condensate through an ordinary heat exchanger before it enters the down leg of the U tube.

As long as some way can be devised for absorbing the excess sensible heat and provided the pressure drop across the plant is small, the U tube provides a beautifully simple method of handling condensate in very large quantities.

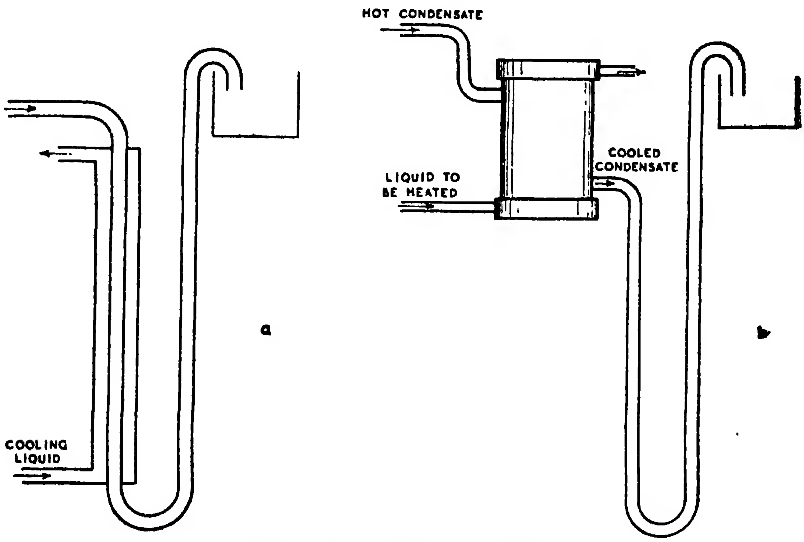


FIG. 153. COOLED U-TUBE

312. PUMPING HOT CONDENSATE. Hot water near the temperature of boiling appropriate to the pressure is always on the look out for a chance to get rid of some heat by flash. If condensate at saturation temperature suffers any pressure drop some steam will flash off. At the suction of a pump, whether centrifugal or reciprocator, there is always a reduction in pressure (that is what suction is) and, unless an extra pressure can be applied to counteract the suction, flashing will occur. Flashing in a pump suction is bad. The pump draws flash steam instead of condensate. The flash steam condenses again under pressure and banging or hammer (or, in a centrifugal pump, a rattling like a lorry on cobbles) occurs. It is essential always to arrange that the natural pressure on hot condensate is augmented by placing the pump suction well below the condensate tank. Table XLIV shows the head, suction or pressure, which has by experience been found satisfactory for water at various temperatures. Where condensate cannot be cooled before pumping, flashing in the pump suction is said to be greatly reduced by the arrangement shown in Fig. 154. Here a heating surface, supplied with steam below atmospheric pressure, cannot be drained by barometric leg owing to lack of headroom. The balance pipe fitted between the pump suction and the heating surface steam space is said to relieve the condensate at the suction of the pump of local drop in pressure. The author has no personal experience of this arrangement and has heard conflicting stories about its efficacy. It would seem more hopeful to provide a large and generous condensate pipe.

313. THE ELIMINATION OF TRAPS. Good traps, well maintained, are satisfactory pieces of plant which function well and give reliable service for years. For really good condensate drainage every heating surface should have its own separate trap. In a large factory this means hundreds of traps. As a trap is small and unobtrusive, working away modestly and generally satisfactorily, maintenance only too often relaxes for more urgent matters. In Sections 307 to 311 it has been shown that in some situations traps can be replaced by piping systems. In Section 305 the correct method of using one large trap in place of a number of small ones has been explained. Every endeavour should be made to reduce the number of traps and to ensure that the remaining large traps are as good as money can buy. If a factory can arrange things so that it has, say, six

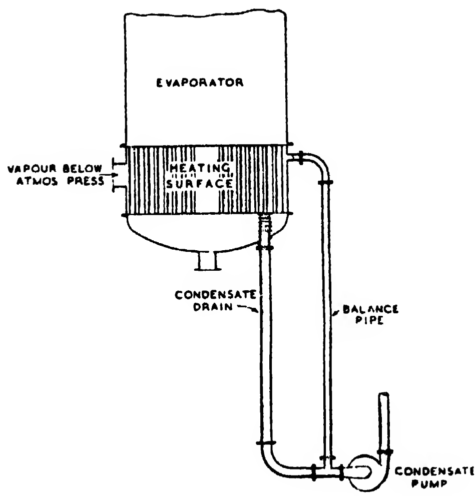


FIG. 154. PUMPING HOT CONDENSATE

TABLE XLIV. SUCTION HEAD FOR PUMPING HOT WATER

(Weir)

TEMPERATURE OF WATER °F.	SUCTION LIFT FEET	PRESSURE HEAD FEET
130	10	
150	7	
170	2	
175	Level	
190		5
200		10
210		15
212		17

large traps instead of sixty small ones, the traps become such important pieces of plant that they will always receive all the needed care. It is, however, no use cutting out traps right and left and introducing all the troubles associated with group trapping. Vacuum systems should always be drained by barometric leg, if there is sufficient headroom. Very low pressure systems can be drained by some form of cooled U tube. The plant in high buildings can and should be group trapped—BUT—the group trapping MUST be correctly done. As far as the author knows there is only one correct way and that way is illustrated in Fig. 144, Section 305.

314. TRAP BYEPASSES. Should every trap be fitted with a byepass ? Answer : Yes, on condition that the byepass is used only by the maintenance staff for examining and servicing the trap. Answer : No, if maintenance is haphazard or non-existent. Answer : Yes, if the trap is draining a high pressure steam main, when prudence may demand that the byepass be wide open during warming up from cold.

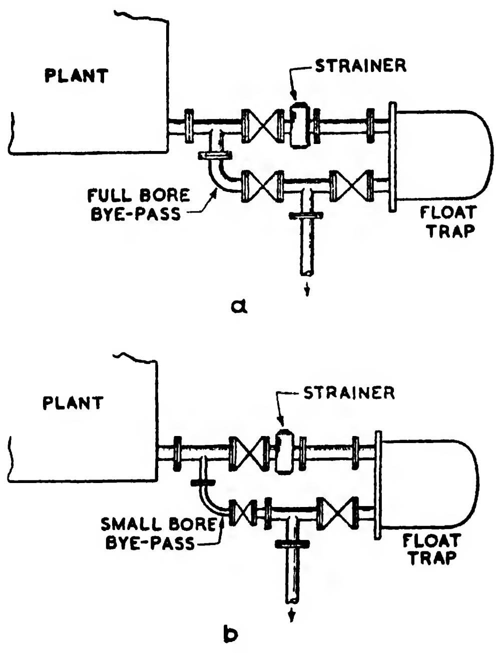


FIG. 155. TRAP BYEPASSES

If heating surfaces get air locked or waterlogged they lose efficiency, output will drop or incorrect processing will occur. At the least provocation a process operator (or the cold occupant of a room) will byepass the trap. Byepassing a trap must show an improved performance, even when the trap was working perfectly. Any air in the system is quickly swept out. Condensate cannot accumulate and is brushed off the surface by the faster moving steam. Heat transfer will be better. How can the operator be blamed for achieving these beautiful effects ?

There is a lot to be said for no byepasses. It means that if a trap chokes with scale or tow, or if it leaks, it will get prompt attention. The operator is unable to compare the performance of his plant with and without the trap in circuit. Provided maintenance is kept well up to the mark all-round performance will probably be better without byepasses.

In the author's factory evolution has resulted in the following rather curious state of affairs. The bulk of the process condensate is dealt with by two huge traps. These are not provided with byepasses. They do not go wrong. If they did, they could be manually operated until an opportunity for servicing came along. The remaining small traps are fitted with byepasses, which are, on occasion, abused. This evolution probably supplies a good answer. Have as few traps as possible. Make them costly and good and they will require no byepasses if they are sufficiently important. Little traps tucked away in odd corners need byepasses which, sooner or later, will be abused.

Trap byepasses should be small, so that the waste when they are opened is as small as possible, and so that there is not a great improvement in performance when the byepass is opened. Byepasses are usually made full pipe size. Fig. 155a shows the usual arrangement. Fig. 155b shows the suggested arrangement where the byepass is made of much smaller piping.

It is a good plan to remove the hand wheel from trap byepass valves so that they cannot be opened by the irresponsible.

In some factories sight glasses can be fitted to trap discharges. These give visible indication of the operation of the trap. These are sometimes satisfactory, but often the condensate quickly coats the glass with a rusty stain, when nothing can be seen. In such cases sight glasses are useless unless regularly and frequently cleaned.

If opening a trap byepass gives a miraculous improvement in performance to a plant that was suspected of being condensate-logged, the condensate outlet should be disconnected from the trap so that the operation of the trap can be observed. From what is seen compared to what should be seen from the particular type of trap a good diagnosis should be easy.

* * *

Whenever condensate above 212° F. is to be handled the greatest care must be taken to avoid water hammer. Therefore all possible precautions must be taken to prevent steam forming by flash and its possible recondensing by coming into contact with cooler water or by increased pressure. Condensate should be handled as expeditiously as possible, by the shortest route with the avoidance of horizontal pipes (condensate pipes should always slope).

There are many other problems connected with the handling of condensate. These concern the collection and use of flash steam and the removal of air. To deal with the whole subject at once would have meant an enormous rambling chapter. Splitting up into separate chapters is by no means satisfactory but has been done—for better or for worse.

* * *

CHAPTER 10

AIR AND ITS REMOVAL

A solemn, strange and mingled air,
'T was sad by fits, by starts 't was wild.
W. COLLINS. *The Passions* 1747.

OF all the problems that confront the steam user, the removal of air from inside heating surfaces is the most neglected, yet it is one of the most important. This chapter will only discuss the problem briefly, but sufficiently to make quite clear the one cardinal rule. For detailed methods of air removal from almost every kind of steam plant Northcroft (see Section 805) should be consulted.

315. AIR AND INCONDENSIBLE GASES. Air may enter plant as air, but the oxygen only too often joyfully attaches itself to part of the plant leaving oxide and rust behind and nitrogen to go on. CO_2 is the constant companion of steam. Water dissolves CO_2 readily and the CO_2 is driven off when the water is heated. Water containing temporary hardness, calcium bicarbonate when heated gives off CO_2 and precipitates calcium carbonate as scale or mud.

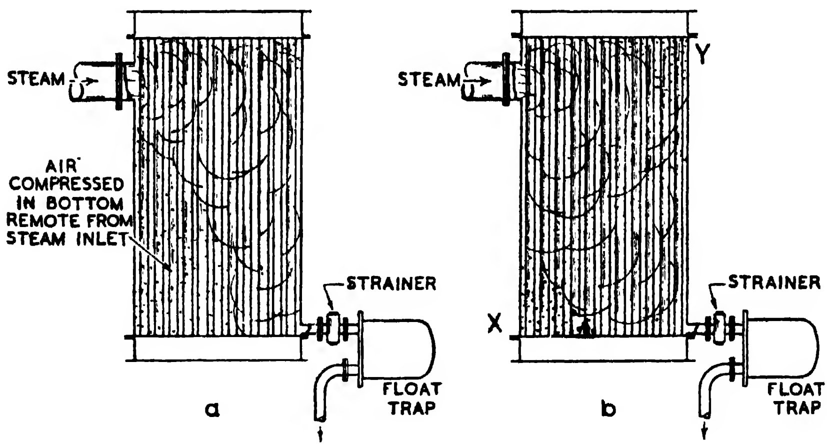


FIG. 156. AIR IN STEAM HEATED CALORIFIER

Many process materials give off gases when heated. In multiple effect evaporation (see Chapter 17) the amount of gas may be considerable. All plant at the start-up is full of air. Every time steam plant is shut down it fills with air sooner rather than later. When the plant stop valve is shut the steam inside the plant condenses and forms a vacuum which is slowly broken by air leaks at joints, valve spindles, glands, etc. All steam, however high grade, contains air or some incondensible gas.

There are two problems in air removal from steam plant. Intermittent plant fills with air at each shut down, and means must be provided for removing

this air as quickly and completely as possible. Continuous plant must be provided with devices which will vent the small amount of air or gas that comes in with the steam.

In the following sections the word "air" means any incondensible gas.

316. RESISTANCE OF AIR FILM TO HEAT TRANSFER. In Chapter 4, Section 162, Table XIX the heat conductivities of various substances are given. It will be seen that air is a worse conductor of heat than the best lagging. A film of air 1/100 in. thick offers the same resistance to heat transfer as does a piece of copper 11 feet thick, a piece of steel 15½ inches thick or a film of water 1/5 in. thick. If it is important to remove condensate it is clearly much more important to remove air which may collect on the heating surfaces.

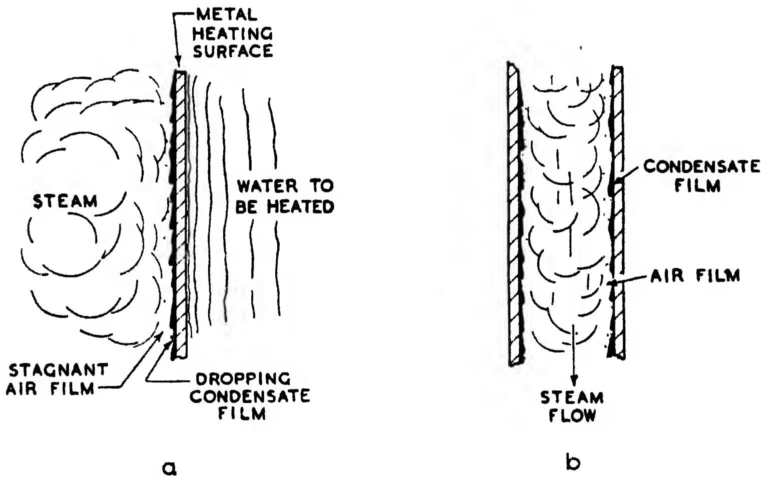


FIG. 157. THE MECHANISM OF CONDENSATION

317. BEHAVIOUR OF AIR IN A STEAM SPACE. Inside a steam space there is always some turbulence. Air can seldom separate out completely from steam. Local convection currents and diffusion help to keep the air and steam partly mixed. When plant is first started up the steam space is full of air. When steam is first turned on the air is compressed and pushed to the far end of the steam space. As air is denser than steam at the same temperature and as, at start up, the air is cooler than the steam, the air will collect at the bottom of the heating surface remote from the steam inlet. Turbulence and diffusion, however, mix the two together so that there is no sharp dividing line. Fig. 156a shows what may be expected to be the state of affairs immediately after starting up a vertical calorifier. If there is no way of escape for the air the state of affairs that may be expected after a few minutes is shown in Fig. 156b. Most of the air has more or less mixed with the steam, but there is more air in the remote corners. "Remote" is a very important word in connection with air removal. It is not measured as Euclid would fly, but as steam would flow. "Remote" is the be-all and end-all of air removal.

Now let us consider a piece of heating surface from which the start-up air has been removed. The plate, Fig. 157a is cool and anxious to take up heat. Steam condenses, and the condensate will run down. Any air in the steam will collect as a stagnant blanket between the condensate and the steam. Heat can now only reach the condensate surface by conduction through the air, or by the steam diffusing through the air. In practice, things are not quite so bad as this. Fig. 157b shows steam inside a heating pipe, with the flow of steam downwards, as it should be. The condensate is flowing downwards and, although the steam is flowing towards the tube walls it does not do so at right angles ; there is a downward movement as well due to the whole body of the steam flowing down the pipe. The air film (imaginary or real) is sandwiched between two downward movers which are almost certainly moving at different speeds. This encourages turbulence and helps diffusion to mix the air back into the steam again, whilst moving the air bodily towards the point remote from the steam inlet.

318. EFFECT OF PARTIAL PRESSURE. What effect has air on steam when it is mixed homogeneously with it? We may look for the answer in Dalton's Law of Partial Pressures, which sounds more alarming than it really is.

> "In a mixture of gases in a given volume, each gas exerts the same pressure that it would exert if it occupied the volume alone. The total pressure exerted by the mixture of gases is equal to the sum of the pressures each would exert if it occupied the volume alone."

Section 9 in Chapter 1 explained that the pressure exerted by a gas was due to the bombardment of the walls of the gas container by the gas molecules. In a mixture of gases, the molecules of each are doing their share of the bombardment—they are each exerting part of the pressure—they are each exerting a partial pressure. So if the law applied to a mixture of one part by volume of air to four parts by volume of steam, the air would exert 1/5 of the total pressure. Then, if the mixture was at 15 psi.a., just above atmospheric pressure, the steam would exert a partial pressure of 12 psi.a. and the air a partial pressure of 3 psi.

Now as the heat in the mixture is provided by the steam, which is at a partial pressure of 12 psi.a., we find from the steam table that the temperature of the mixture must be 202° F.

Dalton's Law holds strictly only for mixtures of so-called perfect gases, see Section 5, but steam is a vapour and the conditions of pressure and temperature are complex. There seems to be little reliable data. Dalton's Law can only help us in plants where the steam is somewhat stagnant in such things as autoclaves or sterilisers, and it is true to say that the temperature will not exceed the temperature appropriate to the partial pressure of the steam up to pressures of about 170 psi.g.

In plant where the steam is enclosed inside a heating surface we really need not worry whether Dalton's law holds or not. In any case, with the steam steadily condensing, local concentrations of air near the heating surface, or in parts of the apparatus remote from the steam inlet, may occur and we have no means of estimating the local partial pressure. Far more important than the actual steam temperature is that the air acts as an insulating blanket on the

heating surface and must be removed to allow the steam molecules free access to the surface. We know from Section 162 that air is the best heat insulator in common use. It is therefore the arch enemy in steam and must be removed as rapidly and completely as possible.

319. KEEPING AIR OUT. With plant that is working intermittently it is virtually impossible to keep air out. Air will leak in past a glanded spindle that appears to be quite tight to water or steam. When the steam condenses at shut down a high vacuum is produced which will suck air in at every pore. We must accept the fact that intermittent plant will always fill with air. We cannot keep it out.

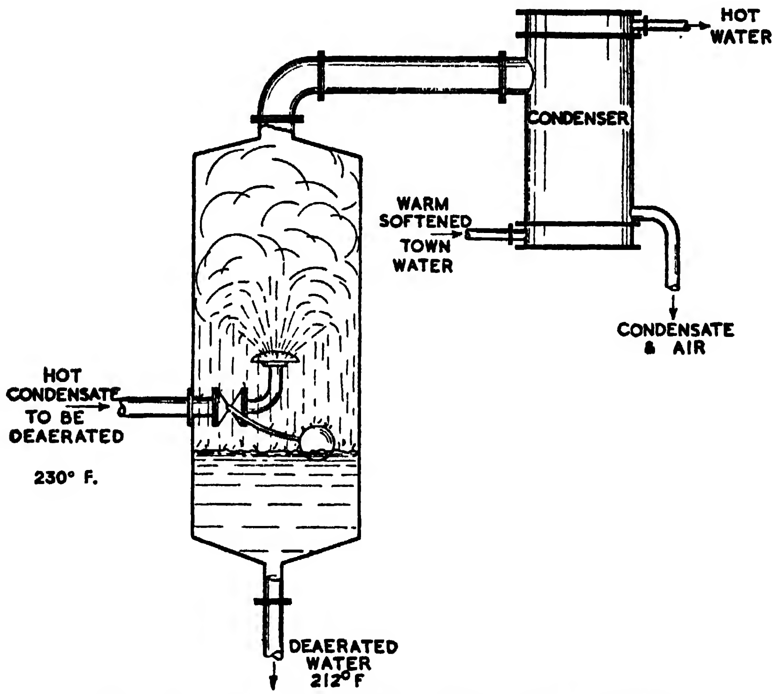


FIG. 158. DE-AERATOR AT LOWERED TEMPERATURE

320. DE-AERATORS. Air in the steam supply is another matter. In all high pressure plants the boiler feed water is de-aerated, but this never completely takes out all the air or CO_2. If there is much air in low pressure steam, it may well pay to de-aerate the feed water. It is important that condensate be de-aerated. Condensate forming inside a heating surface, cheek by jowl with a film of air, has the best of opportunities for saturating itself with air. Condensate may be free from dissolved solids, but it is anything but free from dissolved gases.

The principle of thermal de-aeration is to carry out a limited amount of boiling by flash. This is done by preheating (if necessary) the water and passing it through a flash vessel connected to a condenser and vacuum pump or ejector.

The ejector is very extravagant in steam unless its exhaust steam can be used for heating something usefully. If the water is well above 212° F. all that is needed is to flash the water down to a lower pressure. To get good de-aeration it is necessary to flash down 10° to 20° F. An economic use must be found for the flash steam and, if an ejector is used, for the ejector exhaust.

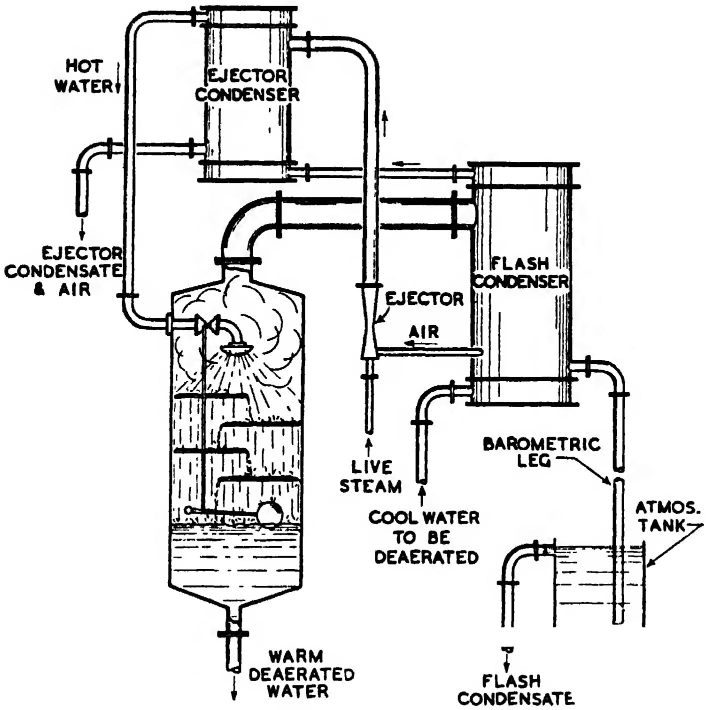


FIG. 159. DE-AERATOR WITH RAISED TEMPERATURE AND HEAT RECOVERY

Fig. 158 shows the simplest form of de-aerator. This can be used where the water is well above 212° F. It requires no description.

Fig. 159 shows an arrangement which supplies external heat to the water, but which returns the heat to the de-aerated water. The cool water to be de-aerated is passed through the tubes of the de-aerator flash condenser where it picks up all the heat that is flashed off in the de-aerator. The warmed water then goes through the tubes of the condenser which condenses the ejector exhaust, which is at about atmospheric pressure. The now hot water is sprayed into the de-aerator, being delayed and broken up in its passage by the perforated trays shown. The vacuum in the de-aerator is produced by the flash condenser and the air is removed by the ejector. The input heat is the live steam to the ejector. This heat is used at a fair temperature in the flashing vessel and reappears, at a lower temperature in the warmed de-aerated water.

It will be seen that in Fig. 158 the de-aeration is effected at a lower temperature than that of the original water, whereas in Fig. 159 de-aeration is carried out at a higher temperature. It is argued by many experts that the method

used in Fig. 158 is unsound because de-aeration by reducing temperature cannot be so complete or effective as by raising the temperature, because the cooler the water the more gas can be held in solution. There may well be something in this argument, but de-aerators of the cooling type shown in Fig. 158 are giving excellent results in the author's factory.

321. METHODS OF REMOVING AIR. There are four ways of removing air from the steam spaces of heating surfaces :—

(*a*) By open vent pipes.
(*b*) By hand operated vent cocks.
(*c*) By automatic means—expansion traps.
(*d*) By using a trap for condensate removal that is inherently air-venting.

The first of these, the open vent pipe, is the best if it can be applied economically. But it can be either very wasteful if too generous or too mean if made too small with a view to economy. Where there are steam mains at a number of different pressures the open vent pipe is ideal. Each heating surface can be pipe vented into the lower pressure main. Venting can be generous because any steam vented with the air will be used by the plant supplied by the low pressure main. The vents must be provided with non-return valves so that when the vented plant is shut down steam from the lower pressure main will not come back into the high pressure plant. The non-return valves are the only things to go wrong. Of course all the air that is vented into the low pressure main has to be vented from the low pressure plant, but the low pressure plant has to be vented anyhow. Open steam blowers heating liquor or water are useful for getting rid of air, and such blowers usually work on very low pressure steam (anyhow they ought to.). It is possible to pick up all the heat lost through open vents in a spray condenser, which is described in Chapter 14.

Hand operated cocks are almost always abused or neglected. When start-up is sluggish the operator will open the vent. When does he shut it ? Too soon, or too late. Once the plant is running the vent is probably not touched again until output drops materially and the operator is reminded by the manager that he has a vent.

Automatic air vents should be the best of all ways of venting, but they are relatively expensive, they must be cared for to see that they are working properly, they can go wrong, but probably most important, they are frequently distrusted. The best automatic air vents are the balanced pressure expansion traps or the liquid expansion traps. The former are lighter, cheaper and have a quicker response but are somewhat fragile. The latter are more robust.

The great advantage of the balanced pressure trap as an air vent (or as a condensate trap) is that its action is differential and is independent of pressure. The liquid expansion trap is a pure thermostat and must be adjusted for a particular pressure at which only will it work. The balanced pressure trap will vent air at any pressure automatically.

Where an inverted bucket or thermostatic trap is used for draining condensate there may be no need to make any separate provision for venting air. Provided the condensate outlet is at the " remote point " of the steam space, the air will be adequately removed by an expansion trap, or, if air removal has not to be done very quickly, by an inverted bucket trap.

322. WHERE IS THE "REMOTE" POINT. In Fig. 156 we saw that the remote point was the stagnant corner remote from the steam inlet, and low down rather than high up. In Fig. 156 the air will probably mostly collect at point X. There may be another stagnant air pocket at Y. Trial and error with vents at various places is the only certain way of finding out. More than one vent is very often necessary. If the steam inlet were low down it is almost certain that a vent should be fitted near the top as well as near the bottom.

A few examples of air venting will now be given. In each case it will be assumed that, for some reason, it is necessary to use a condensate trap that is not inherently air-venting.

In Fig. 160 (compare Fig. 144) the air will be swept along with the condensate to the end of the coil. Some will be entangled in the condensate and will be carried into the trap tank, which of course must be vented. But there is no real driving force to send it down the condensate pipe. Air vents should therefore be fitted at the top of the condensate pipes.

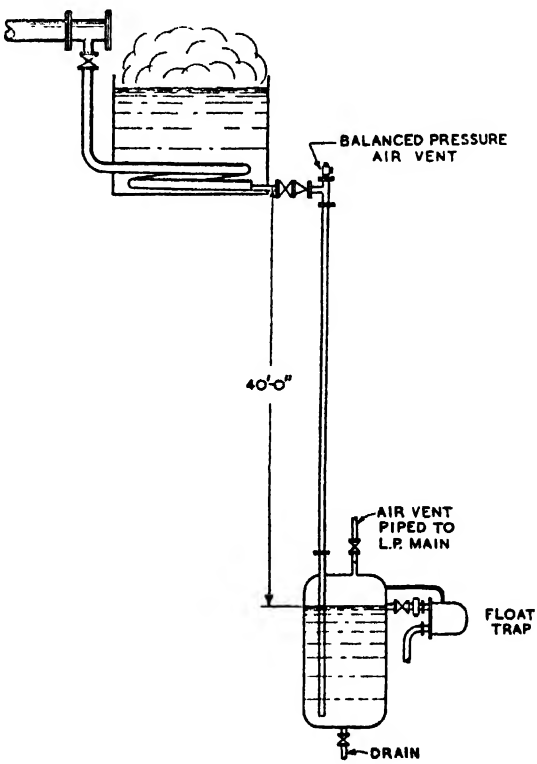


FIG. 160. AIR VENTING OF TANK COIL

In Fig. 161 (compare Fig. 145) the air will be driven and will fall to the bottom of the heater and will collect in the trap. The trap should be fitted with a small balanced pressure expansion trap. In Fig. 145b the balanced pressure trap will remove all the air and there is no need to fit any special device for venting.

In Fig. 162a (compare Fig. 140) the top of the jacket is clearly the remote point and the vent should be fitted there. But in Fig. 162b the remote point is at the bottom—the right place—and the vent should be attached to a small pipe extending into the middle of the jacket bottom, well above the condensate level.

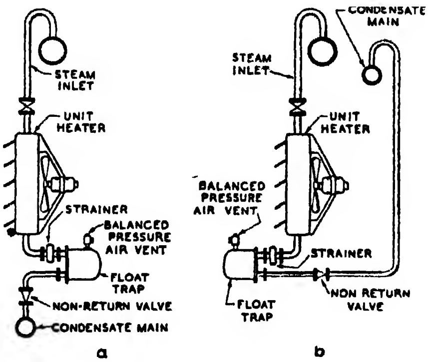


FIG. 161. AIR VENTING OF UNIT HEATERS

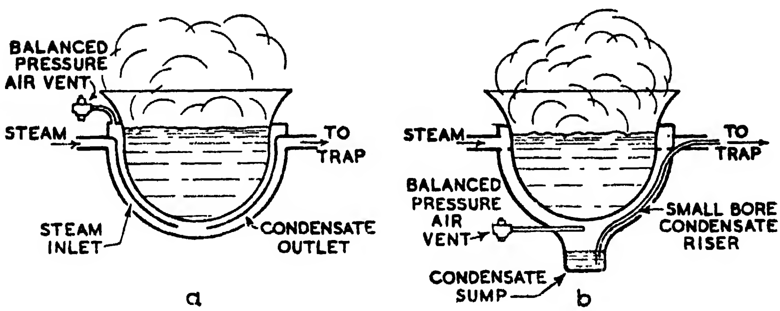


FIG. 162. VENTING HEMISPHERICAL JACKETED PAN

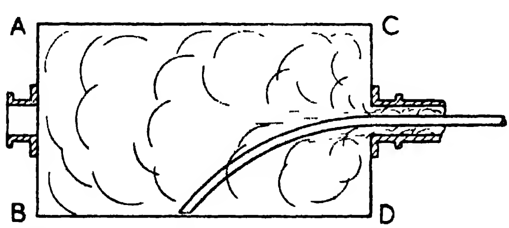


FIG. 163. VENTING PROBLEM OF DRYING CYLINDER

The drying cylinder, Fig. 163 again sets the difficult problem. Where is the remote point ? Possibly A or B, more probably C or D, perhaps all four. Now inside a drying cylinder things are pretty dull. The volume is huge and the heat transfer is relatively small. The steam movement is very sluggish

and cannot be relied on to sweep the air to any particular point. Steam circulation, which is dealt with in Chapter 18, is a possible way out, because it moves the air into a place where it can separate and be vented. Air vents should be attached to the cylinder certainly at C or D and probably at A or B as well. The vents should be as small as possible because clearances are usually small. If there is not enough clearance to allow automatic vents to revolve with the cylinder, plain open vents should be fitted.

Wherever possible a foot or two of bare pipe should separate an automatic vent from the plant being vented. This allows the air to cool and permits the venting trap to work with more certainty.

When steam locking occurs and the condensate trap is fitted with a steam lock release there is no need to fit a separate air vent provided that the condensate is piped off at the remote point.

Even if good air vents are fitted there is always a chance that air may get into a trap, which, if not inherently air venting, will eventually lock. In such cases it is not necessary to fit a special air releasing trap to the trap; all that is needed is to connect a small pipe from the top of the trap back to the vented steam space. See Figs. 160 & 164.

The essence of air venting is the finding of the remote point. Better still is to make the steam follow a certain path to a prearranged remote point. Such a system of deliberately forcing the steam to drive the air to a remote point is shown in Fig. 165. This is an evaporator calandria provided with two hexagonal baffles so that the steam has a definite path. Not only is this beneficial for driving the air into a corner, but it increases the steam velocity over the tubes and helps to brush off the condensate.

323. VENTING VACUUM SPACES. Steam at less than atmospheric pressure is condensed inside condensers and inside the heating surfaces of some plants, for example, in multiple effect evaporators. The air in the steam will collect at the remote point, as in pressure plants. An open vent, or a thermostatic vent to atmosphere will not let the air out; either would let more in. The air must be extracted. In the heating surfaces of an evaporator this is easy. An open vent from the heating surface into the vapour space well above the liquid is all that is needed. See Fig. 165. This of course is wasteful, but it works. In some cases a thermostatic air vent can be used in such conditions provided the discharge is into the body of the evaporator.

The venting of a jet condenser hardly needs mention. The outlet to the vacuum pump is the vent. It is the remote point and the coolest point—ideal. See Fig. 165.

The vent point in a surface condenser is also the remote point and is connected to the vacuum pump or ejector. See Fig. 166. In the case of surface condensers the remote point is not left to chance. It is carefully arranged by a system of baffles.

324. VACUUM BREAKING—THE ADMISSION OF AIR. It is important that steam pipes should be prevented from sucking back liquids through leaky coils when the plant is shut down at night or at the week-end.

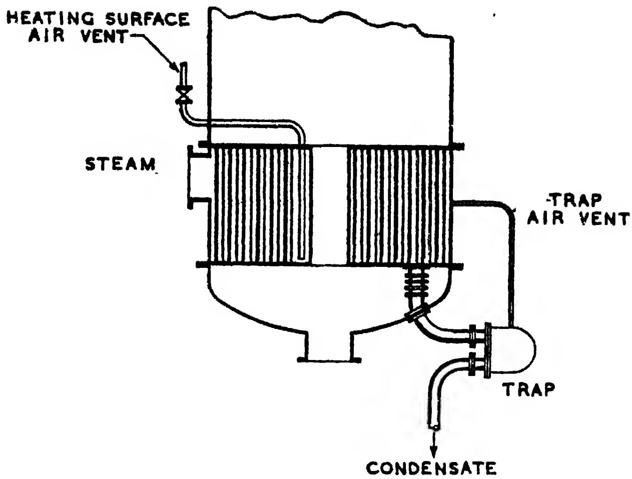


FIG. 164. VENTING FLOAT OR OPEN BUCKET TRAP

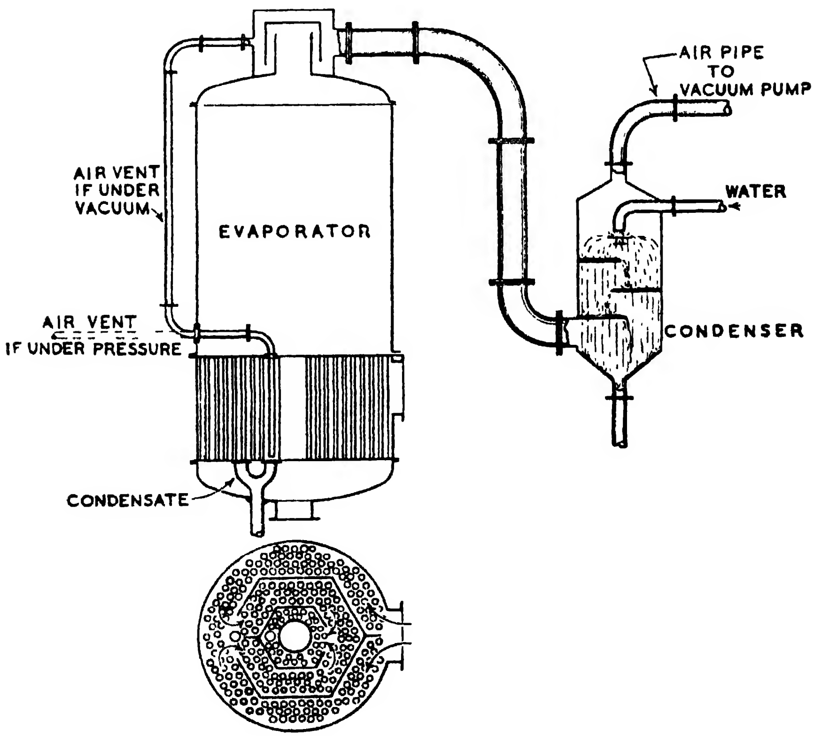


FIG. 165. CALANDRIA PAN WITH AIR REMOVAL FROM CALANDRIA. CALANDRIA BAFFLED TO SWEEP AIR AND CONDENSATE TO VENTING AND DRAINING POINTS. VAPOUR IS CONDENSED BY A TRAY TYPE JET CONDENSER

(87664)

Normally at the shut down all steam valves are closed. When the plant cools the steam condenses in the pipes and forms a vacuum. This frequently results in drawing liquid from tanks and vats back through minute pinholes or cracks into coils. When the plant is started up this liquid is discharged through the trap into the condensate line and may contaminate many thousands of gallons of what should be distilled water. The risk of this happening is very great where the steam coil is heating a very corrosive liquid which may easily perforate the coil—in electro-plating, for example.

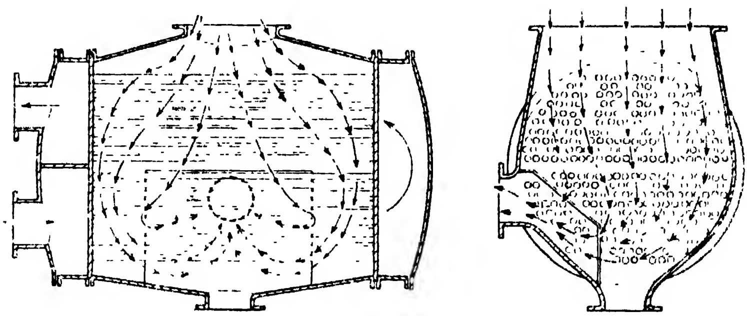


FIG. 166. AIR SWEPT TO ONE POINT IN SURFACE CONDENSER

Occasionally the liquid is drawn right back into the steam pipe from leaky coils past leaking valves. Automatic vacuum breakers can be fitted, though they can stick shut. It may be better to drill a pinhole in the steam pipe between the valve and the coil. This is admittedly wasteful, but it is simple and reliable.

* * *

Air venting is of the greatest importance. It often greatly increases output. By increasing the temperature of the steam, it allows a lower pressure steam to be used. Northcroft gives some striking figures ; textile drier heating-up time reduced from 90 to 10 minutes by proper venting ; jam boiling time reduced by 30 per cent. and the steam consumption reduced by 20 per cent. by proper venting. In the author's factory the first body of a double effect evaporator mysteriously dropped from 18 in. vacuum to atmospheric pressure due to the vent on the second calandria having been inadvertently closed.

Air venting is simple. It only requires patience and enough imagination to determine the "remote" point. Every steam heating surface must be provided with proper means for venting the air. The vent must be at the remote point. If in doubt, fit more than one vent. Try to arrange the flow of steam inside the heating surface so that the air is definitely swept to one point.

The brevity of this Chapter is no indication of the importance of the subject. The subject is so important that a whole book would be needed to deal adequately with all the possible plants. Northcroft provides the whole book. But sufficient has been said to enable anyone to look for the remote point and vent it.

CHAPTER 11

HEAT TRANSFER BY HEATING SURFACE

. . . One heat another heat expels.
SHAKESPEARE. *Two Gentlemen of Verona.* 1623.

STEAM heat is transferred in two ways : through a heating surface or by direct contact. The mechanism of the molecular movements that bring about the heat transfer from the steam to the product have been described in Chapter 1, Sections 15 and 17.

325. HEATING SURFACES AND DIRECT CONTACT. A heating surface is a wall with the substance that is to be heated on one side and the heat-providing medium on the other. The object of the heating surface is to provide a separator which will pass heat, but will not allow the substances on either side of it to mix. In most heating or cooling operations, it is quite inadmissible to allow such mixing. It clearly would be inadmissible to condense steam with dirty canal water in a jet condenser if the condensate were required for boiler feed. Another reason for keeping heater and heated separate is to enable the heating medium to have a higher temperature than the heated product.

There are, nevertheless, a number of applications where direct steam heating can be used. The heating of water is by far the most important. The heating of aqueous solutions that are dissolving solids or are being diluted, and the special application of the steam distillation of certain high boiling point substances are other common uses for direct steam.

Direct contact calls for the simplest possible plant. Heating surface plant is much more costly and much larger, and heating surfaces possess inherent drawbacks which are not always easy to circumvent. However, the maintenance of direct contact plant is sometimes more tiresome and more expensive than the maintenance of heating surface plant.

326. HEAT TRANSFER RESISTANCE. Heat conductivity and electrical conductivity are very similar and obey the same laws. Conductance is a particular value taking account of thickness or length as well as the physical properties of the material. The specific conductance or "conductivity" is the conductance per unit length or unit thickness per unit temperature increase in unit time. The resistance to heat or to electrical flow must vary directly with the length or thickness of the material and inversely as the conductivity.

$$\text{Resistance} = \frac{\text{Thickness}}{\text{Conductivity}} \qquad R = \frac{L}{\lambda}$$

The resistance to heat flow through a heating surface is built up of several resistances : the condensate film, any air blanket, the metal wall itself, scale, etc. In just the same way the resistance of an electrical circuit is built up of the generator resistance, the feeders, the distribution wires, the motor resistance, etc.

In an electrical circuit the total overall resistance is found by adding together the component resistances. Similarly, in heat flow, we can find

the overall resistance to heat transfer by adding together the individual resistances.

$$\text{Overall Resistance} = \frac{\text{Thickness of metal}}{\text{Conductivity of metal}} + \frac{\text{Thickness of condensate film}}{\text{Conductivity of water}} + \text{etc.}$$

$$R = \frac{L_1}{\lambda_1} + \frac{L_2}{\lambda_2} + \frac{L_3}{\lambda_3} + \frac{L_4}{\lambda_4} + \text{etc.} \quad \ldots \quad (a)$$

Now conductance and resistance are reciprocals, so that

$$\text{Conductance} = \frac{1}{\text{Resistance}} \quad \ldots \quad (b)$$

For heat flow problems we want to know the overall conductance, that is the heat that will flow, not the resistance offered to flow. We find this by combining the two formulae (*a*) and (*b*) just given.

$$\text{Conductance} = U = \frac{1}{R} = \frac{1}{\frac{L_1}{\lambda_1} + \frac{L_2}{\lambda_2} + \frac{L_3}{\lambda_3} + \frac{L_4}{\lambda_4} + \text{etc.}}$$

The conductance U is called the Overall Heat Transfer Coefficient or *Heat Transfer Rate.* As the conductivity is expressed as so many Btu/sq. ft./hr./° F. diff./unit thickness, the Heat Transfer Rate will be measured in Btu/sq. ft./hr./° F. diff.

Let us now examine the resistance offered to heat flow by an imaginary heating surface. For convenience part of Table XIX is repeated below—Annexe A. Let us assume that we have a heating surface made of steel ¼ in. thick, in a steam calorifier heating water.

ANNEXE A (FROM TABLE XIX)

MATERIAL	BTU/SQ. FT./HOUR/°F. DIFF./INCH THICKNESS
Copper	2,620
Steel	310
Water	4
Scale	1 to 12
Air	·2

Let us assume that there is a film of condensate 1/100 in. thick on the steam side, and that this film is blanketed by a film of air also 1/100 in. thick. (Actually this will not be a true " film " of air. Diffusion of air molecules into the steam and steam molecules into the air is very rapid. There will be an increasing concentration of air nearer the heating surface. For pointing the heat transfer lesson, however, it will be assumed that there is an actual air film.) Let us also assume that there is 1/100 in. of scale on the water side and a stagnant film of water 1/100 in. thick between the scale and the body of the water. (In addition to these films there may be another and very resistant

film in evaporators if a film of steam forms on the liquid side of the heating surface, see section 348.) We can tabulate this information thus :—

Item	*Thickness*	*Conductivity*
Air	$L_1 = \cdot 01$	$\lambda_1 = \cdot 2$
Condensate	$L_2 = \cdot 01$	$\lambda_2 = 4$
Steel	$L_3 = \cdot 25$	$\lambda_3 = 310$
Scale	$L_4 = \cdot 01$	$\lambda_4 = 2$
Water film	$L_5 = \cdot 01$	$\lambda_5 = 4$

The Heat Transfer Rate will be :—

$$\frac{1}{\frac{\cdot 01}{\cdot 2}+\frac{\cdot 01}{4}+\frac{\cdot 25}{310}+\frac{\cdot 01}{2}+\frac{\cdot 01}{4}} = 16\cdot 45 \text{ Btu/sq. ft./hr./° F. diff.}$$

327. EFFECT OF USING COPPER INSTEAD OF STEEL. The conductivity of copper is 2,620 against 310 for steel, so that λ_3 will equal 2,620. The Heat Transfer Rate will be :—

$$\frac{1}{\frac{\cdot 01}{\cdot 2}+\frac{\cdot 01}{4}+\frac{\cdot 25}{2{,}620}+\frac{\cdot 01}{2}+\frac{\cdot 01}{4}} = 16\cdot 64$$

So that, although copper is more than 8 times as good a conductor as steel, and although the heating surface is ¼ in. thick, the effect of using copper instead of steel is quite negligible in this example. This is because the resistant films, so long as they are substantial, are so much more important that the thickness or material of the heating surface do not come into the picture. When great pains have succeeded in reducing the films it may pay to use copper or brass instead of steel. The point that is important is that the metal is the least important.

This broad statement that the metal of the heating surface is of little importance must be accepted with reserve. In Section 348 a state of affairs is discussed where a great increase in heat transfer rate can be sometimes secured by using non-ferrous heating surface. But this effect has nothing to do with the heat transfer through the metal, it is a surface effect which increases the rate of heat transfer from the steam into the metal.

328. EFFECT OF SCALE. Scales vary in composition and the published figures of the resistance of scale to heat transfer are very conflicting. Apart from the effect of the material composing the scale, other effects may be introduced. The tabulation overleaf shows the week's reduction in Heat Transfer Rate of two evaporators in the author's factory. These evaporators are descaled each week-end. The material being evaporated is an impure sugar solution entering with 85 per cent. water and leaving after concentration with 30 per cent. water. The evaporators are continuous, not batch.

It is almost impossible to measure the thickness of the scale which is deposited much more heavily in certain parts of the tubes. Weighing the amount of scale removed gives the average thickness as $\cdot 0012$ in., if spread evenly over the heating surface. (The scale is actually confined almost entirely to the first effect.)

	Double Effect Short Tube Evaporator	Double Effect Long Tube Climbing Film Evaporator	Average of two Evaporators
		Btu/sq. ft./hr./°F. diff.	
Monday	260	280	270
Tuesday	210	260	235
Wednesday	200	220	210
Thursday	190	220	205
Friday	160	180	170

We can try to calculate the conductivity of the scale thus :—

If x = the overall resistance of the clean tube and any air or liquid films and y = the conductivity of the scale

$$\frac{1}{x} = 270 \qquad \therefore x = \frac{1}{270}$$

$$\frac{1}{x + \frac{\cdot 0012}{y}} = 170$$

$$\therefore \quad \frac{1}{\frac{1}{270} + \frac{\cdot 0012}{y}} = 170$$

$$\therefore \quad y = \cdot 55 \text{ Btu/sq. ft./hr./° F temp. diff./Inch thickness.}$$

This is a lower conductivity than the published figures. The scale composition is not very unusual, it is a mixture composed chiefly of calcium sulphate with varying proportions of calcium carbonate and calcium phosphate.

It is possible that the effect of the scale is not simply to insulate the heating surface, but by so doing to upset the circulation of the evaporator (see Sections 366 and 367 in the next Chapter). It was also known that all the scale that was removed was not weighed.

329. EFFECT OF REDUCING RESISTANT FILMS. We can work out all kinds of combinations of film thicknesses and tabulate the results, as in Table XLV.

It is a dull job reading a table, but Table XLV does deserve reading and study because it contains the whole story of heat transfer through a heating surface. With films such as may be quite common if no precautions are taken to reduce them, it does not matter in the slightest what the thickness of the heating surface is or of what material it is made. Halving the thickness of the water films or scale makes precious little difference while there is still the equivalent of 1/100 in. of air. Halving the air blanket but leaving the other films untouched increases the heat transfer rate in the ratio 3 to 5. Until the air and the scale have been practically eliminated, there is little use in going to the expense of copper. Even when air and scale are eliminated, halving the

thickness of a copper heating surface makes practically no difference. It is not until *all* the resistances have been tackled, removed or greatly reduced, that the material or thickness of the heating surface has any say in the heat transfer rate. So that is why we have had a chapter on air removal, a chapter on condensate removal and are going to have two part chapters on material movement.

TABLE XLV. HEAT TRANSFER RATES WITH VARIOUS RESISTANT FILMS

HEATING SURFACE		CONDENSATE FILM THICKNESS	NOTIONAL AIR FILM THICKNESS	SCALE THICKNESS	WATER FILM THICKNESS	OVERALL HEAT TRANSFER RATE
METAL	THICKNESS	INCH	INCH	INCH	INCH	BTU/SQ. FT./HR./°F. DIFF.
Steel	·25	·01	·01	·01	·01	16·44
Copper	·25	·01	·01	·01	·01	16·64
Copper	·01	·01	·01	·01	·01	16·67
Steel	·25	·005	·01	·01	·01	16·79
Steel	·25	·01	·01	·005	·01	17·15
Steel	·25	·01	·005	·01	·01	29·93
Steel	·25	·005	·005	·005	·01	31·20
Steel	·25	·01	·01	—	·01	17·9
Steel	·25	·01	—	·01	·01	92·6
Steel	·25	·01	—	—	·01	172
Steel	·25	·005	—	—	·005	302
Copper	·25	·005	—	—	·005	385
Copper	·125	·005	—	—	·005	391
Copper	·0625	·0025	—	—	·0025	790
Steel	·25	—	—	—	—	1,240
Copper	·25	—	—	—	—	10,520

330. TEMPERATURE DROP. Drop in temperature in a heating process follows the same law as drop in voltage in an electrical circuit. In an electrical circuit the voltage drop splits itself up and is directly proportional to the resistance of each component. Similarly, the total heat drop across a heating surface will be split up into separate drops across the individual films in proportion to the resistance of each film. Continuing the electrical analogy, if we know the heat transfer rate of a surface and know the amount of heat we wish to transfer we can find the temperature drop that will be required.

Let us take the example given in Section 326.

Material	*Thickness*	*Resistance*	*° F. Drop per ° F. Total Drop*
Air film	·01 in.	·05	·8223
Condensate film	·01 in.	·0025	·0411
Steel	·25 in.	·000807	·0133
Scale	·01 in.	·005	·0822
Water film	·01 in.	·0025	·0411

Now suppose the steam is at 15 psi.g. it will have a temperature of 250° F.; and suppose the water to be heated is at 200° F. The total temperature drop is 50° F. and it will be distributed as follows :—

Across air film	41·1° F.
Across condensate film	2·0° F.
Across steel	·7° F.
Across scale	4·1° F.
Across water film	2·1° F.
	50·0° F.

The temperatures at the various points will be :—

Between steam and air	250° F.
Between air and condensate	208·9° F.
Between condensate and metal	206·9° F.
Between metal and scale	206·2° F.
Between scale and water film	202·1° F.
Between water film and water	200° F.

Fig. 167a shows an imaginary enlarged cross section of this heating surface and shows strikingly the effect of these resistant films. The full line shows the temperature drops given above. The dotted line shows what the temperature drops would be were the air and the scale eliminated. This shows that the same heat transfer would be obtained with a 4·8° F. temperature drop instead of 50° F. Instead of requiring steam at 15 psi we could use vapour at 4 in. vacuum and get the same result. So we learn that by reducing or eliminating air, scale, condensate and stagnant heated material films we do any of the following useful things : use smaller plant ; get greater output from the same plant ; work at lower temperatures ; shorten processing time ; use lower pressure steam. All these things are so desirable that we must strive for them in every possible way.

In practice air cannot exist as a homogeneous film. Ordinary diffusion, due to the rapid molecular movement, quickly mixes the air into the steam and the steam into the air. Acting in opposition to this is the movement of the condensing steam towards the heating surface.

Assume we have steam at 20 psi.g. inside the heating surface of a plant that is to boil water under atmospheric pressure and assume that the Heat Transfer Rate (see next Section) is 150 Btu/sq. ft./° F. diff./hr.

The temperature of saturated steam at 20 psi.g. is 259° F.

So that the temperature difference will be 47° F.

The total heat transfer will be

$$150 \times 47 = 7{,}050 \text{ Btu/sq. ft./hr.}$$

The latent heat at 20 psi is 940 Btu/lb.

$$\text{that the weight of steam condensed} = \frac{7{,}050}{940} = 7\cdot5 \text{ lb./sq. ft./hr.}$$

The volume of steam at 20 psi is 2·0 cu.ft./lb.

So that the volume condensed will be

$7·5 \times 12 =$ 90 cu. ft./sq. ft./hr.

The velocity with which the steam will flow towards the heating surface will be 90 ft./hr.
or 18 in./min.
or ·3 in./sec.

Now air in such a steam space should diffuse into the steam over $\frac{1}{2}$ in. of distance at about ·3 in./sec.

Immediately air blankets the heating surface condensation slows down and the rate of diffusion will approach the rate of steam flow towards the surface. This will allow steam molecules to reach the surface, but to do so they have to shoulder their way through a mass of useless passengers—the air molecules.

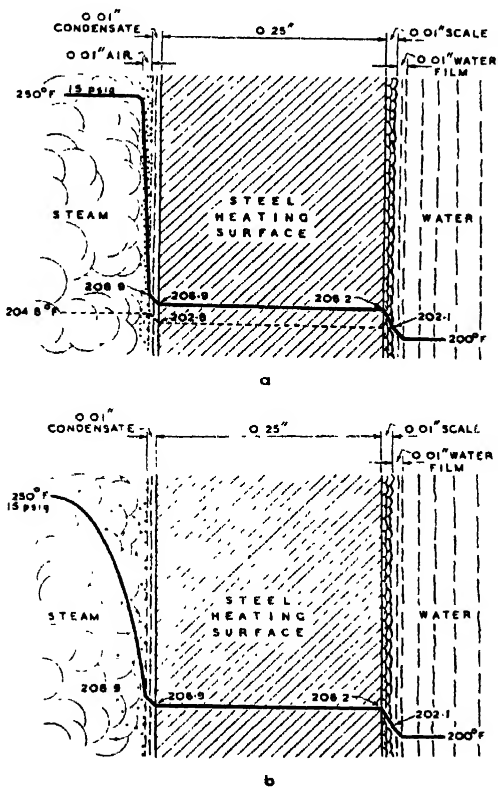


FIG. 167. TEMPERATURE DROP ACROSS HEATING SURFACE AND ITS ADHERENT HEAT-RESISTANT FILMS

Fig. 167b shows the more likely temperature drop. As the air concentration increases nearer the heating surface, so the resistance to heat transfer increases and the temperature curve drops more steeply.

Very little information seems to be available as to the actual effect of air in steam on actual Heat Transfer Rates. Some results published by the American Institute of Chemical Engineers in 1935 gave the following effect of air on the Heat Transfer Rate of the steam side only.

Per Cent. Air by Weight	*Per Cent. H.T.R. Steam side only*
0	100
0·5	87
1·0	76
1·5	71
2·0	69
2·5	68
3·0	67

If these figures are right it means that it is of the utmost importance that the last trace of air be removed.

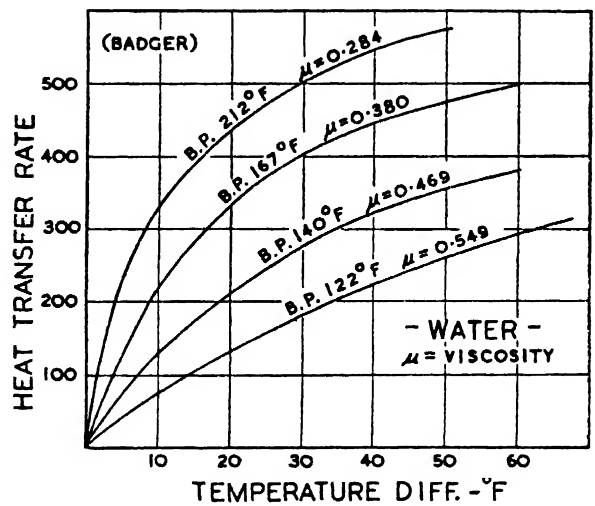


FIG. 168*. VARIATION OF HEAT TRANSFER RATE WITH VARYING TEMPERATURE DIFFERENCES AND VARYING TEMPERATURES

331. OVERALL HEAT TRANSFER RATE. The separate estimation of the constituents of the resistance to heat flow is seldom practicable, and the published data is meagre and conflicting. The only thing that matters is the overall rate, and provided we bear constantly in mind that it is the films that matter we can set about increasing the overall rate. Tests are always done on the overall rate and there is a certain amount of published data. This data is unsatisfactory. From Table XLV it might be thought that the overall heat transfer rate would lie in the tens. Fortunately in most liquid heating problems

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this is not so. It is possible so to reduce the films as to bring the rate into the hundreds. Laboratory results have even reached the thousands.

Heat transfer rate per ° F. of temperature difference not only increases with higher average temperature but increases with greater temperature differences. It seems that the principal reason for the improvement at higher temperature is due to a reduction in viscosity of the heated liquid, thus making the liquid flow more easily and the liquid films thinner.

Fig. 168 shows the effect of increased temperature difference and increased actual temperature on boiling water, as determined by Badger on a semi-full scale laboratory plant.

For space heating and low temperature drying, the published figures seem to show that heat transfer rate to air appears to vary almost directly with temperature drop. Temperature does not seem to be so important with air as with liquids.

Table XLVI gives a few figures collected from various sources. Different observers give quite different figures for the same operation on similar plant. This, really, is only to be expected. 1/1000 in. of air would make all the difference and might well halve or double the figure.

TABLE XLVI. PUBLISHED OVERALL HEAT TRANSFER RATES

PLANT	CONDITIONS	BTU/SQ. FT./HOUR/°F. DIFF.
Condenser	Water flowing at 1 ft./sec.	150 to 400
Condenser	Water flowing at 2 ft./sec.	200 to 550
Condenser	Water flowing at 4 ft./sec.	250 to 750
Boiler	Gas to water	2 to 8
Economiser	Gas to water	1 to 5
Superheaters	Gas to steam	2 to 6
Tank coils	Temp. difference 50° F.	100 to 225
Tank coils	Temp. difference 100° F.	175 to 300
Tank coils	Temp. difference 200° F.	225 to 475
Air heater	Convection only	2
Air heater	Air velocity 1 ft./sec. Temp. diff. 100° F.	3
Air heater	Air velocity 5 ft./sec. Temp. diff. 100° F.	5
Air heater	Air velocity 10 ft./sec. Temp. diff. 100° F.	8

From Table XLVI we can learn several important things. Heat transfer in practice is largely an experimental art. The higher the temperature at which the operation is carried out the higher the rate of heat transfer. The higher the velocity of the heated material the higher the heat transfer rate.

From the rather foggy field of Table XLVI and from certain other figures we will back our fancy and produce Table XLVII.

Fortunately we steam users can pass the onus of designing heat transfer plant on to the manufacturers who, by long experience, have a store of data which enables them to foretell heat transfer rates for various materials and

(87664) K*2

applications with fair certainty. The important thing that the user must always remember is that the working of any heat transfer plant is dependent on the elimination or reduction of the resistant films.

TABLE XLVII. REASONABLE PRACTICAL HEAT TRANSFER RATES

OPERATION (ON WATER OR DILUTE LIQUORS)	BTU/SQ. FT./HOUR/°F. DIFF.
Calorifiers and condensers with low velocity liquid	150
Calorifiers and condensers with high velocity liquid	300
Tank coils, low pressure with natural circulation	100
Tank coils, high pressure and natural circulation	200
Tank coils, low pressure with assisted circulation	200
Tank coils, high pressure with assisted circulation	300
Natural circulation evaporators with low pressure steam ..	300
Natural circulation evaporators with high pressure steam ..	500
Assisted circulation evaporators	750
Space heating by water—convection only	1·5
Space heating by steam—convection only	2
Space heating by steam—10 ft./sec. air velocity	8

332. HEAT TRANSFER RATE AND TOTAL HEAT TRANSFER. In the next 40 Sections the effect of various things on Heat Transfer Rate will be discussed. On many occasions it will be said that this or that decreases the Heat Transfer Rate. That does not necessarily mean that the Total Heat Transfer has been reduced. Heat Transfer Rate is the Heat Transfer from a unit area, in a unit time per degree of Temperature Difference.

It will be stated later that the effect of operating at a lower temperature, in say an evaporator, is to reduce the Heat Transfer Rate. But by reducing the boiling temperature in the evaporator the temperature difference may have been increased. The Total Heat Transfer may therefore be unchanged, or may even be increased.

Look back at Fig. 168, Section 331. Suppose we are evaporating a liquid which is boiling at 212° F. under atmospheric pressure and that the heating is being done by 5 psi steam. The temperature difference is $227 - 212 = 15°$ F. The top curve of Fig. 168 shows that the Heat Transfer Rate will be about 395, and the Total Heat Transfer will be $395 \times 15 = 5{,}925$ Btu/sq. ft./hour.

If now we do the evaporation under the modest vacuum of 18·5 in., the liquid will boil at 167° F. If the temperature difference is kept constant at 15° by using vapour at 13·5 in. vacuum for heating the evaporator, it will be seen from the second curve in Fig. 168 that the Heat Transfer Rate will drop to 280 and the Total Heat Transfer will be $280 \times 15 = 4{,}200$ Btu/sq. ft./hour. If however 5 psi steam is retained for heating, the temperature difference will be $227 - 167 = 60°$ and the Heat Transfer Rate will be 500. The Total Heat Transfer will be $500 \times 60 = 30{,}000$ Btu/sq. ft./hour.

333. IMPORTANCE OF MOVEMENT. Fig. 167 shows most strikingly that we need not worry much whether our heating surface is 1 in. thick or $\frac{1}{8}$ in. thick or whether it is made of steel or copper, until we have dealt with the films. The air film—the most resistant—can be almost eliminated by giving the steam a good quick path to a remote venting point. The condensate film is greatly reduced by good design, not only to enable the film to run freely down and away but to give the steam every chance to brush the film along. Steam circulation, dealt with in Chapter 18, helps these things and sometimes has useful applications. Movement of the material to be heated is generally very important and is often not given the attention it deserves.

In Fig. 167 the stagnant liquid film is taken to be water 1/100 in. thick. It might be tar or molasses $\frac{1}{8}$ in. thick. Data as to the conductivities of materials other than water or oil are extremely scarce and what there are are often conflicting. If the material is very viscous it is probable that the stagnant material film will be the largest component of the resistance opposing heat transfer. Fig. 169

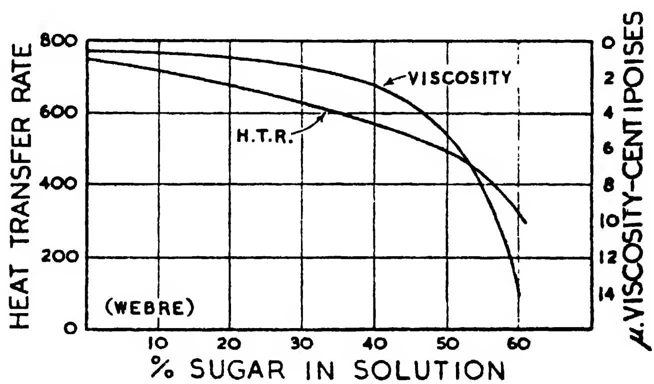

FIG. 169*. VARIATION OF HEAT TRANSFER RATE WITH VISCOSITY IN SUGAR SOLUTIONS

shows the relation between heat transfer and viscosity of the less viscous sugar solutions. Most of the sugar solutions dealt with by the sugar refiner are denser than those shown in Fig. 169, where it can be seen how rapidly the viscosity rises, and the heat transfer falls, with increased density.

Viscosity can only be reduced by increase of temperature or by dilution. High temperatures may be technically impossible. Many organic liquids will not tolerate high temperatures. The process may be evaporation or concentration—the reverse of dilution. In many cases then, we just have to make the best of viscosity and the stagnant film must be dragged along, torn off, mixed up. This can only be done by giving the material as much movement as possible, by making the movement easy, by avoiding stagnant corners. With very viscous materials it is of little use relying on natural circulation or flow. Some industrial process materials have a consistency not unlike sausage-meat coupled with a stickiness beyond the dreams of small boys.

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334. MOVEMENT IN CALORIFIERS AND HEAT EXCHANGERS. A calorifier or heat exchanger is essentially a large heated pipe. The movement is just the flow through the vessel. If there is sufficient head, either static head or head from a pump, the velocity through the heat exchanger can be made within limits, as high as we like. Suppose the calorifier has 2 in. tubes and each tube is 7 ft. 6 in. long, each tube will have a capacity of one gallon. Suppose 96 gallons per minute of liquid are to be heated and there are twelve tubes in one single bank, the rate of flow will be 1 ft./sec. If the vessel is now changed to double pass, with 6 tubes in each pass, the flow will be increased to 2 ft./sec. With three passes of 4 tubes each, the velocity would be 3 ft./sec. ; with 4 passes of three tubes each, 4 ft./sec., and so on until the limit is reached at 12 passes of one tube each giving 12 ft./sec.

What improvement in heat transfer can be expected by an increase in liquid velocity ? This depends on the type of flow through the pipe or tube. Flow can be of three types : "Streamline", "Turbulent" or "Transitional". The type of flow is determined by the "Reynolds Number" (R_e). This sounds much more daunting than it really is. The flow of fluids was studied by Osborne Reynolds who devised his "Number" in 1875. The Reynolds Number takes account of the velocity of flow, V in ft./sec., the pipe diameter, D in feet, the density of the liquid, ρ in lb./cu. ft., and the viscosity, μ in lb./ft. sec. units.

$$\text{The Reynolds Number } R_e = \frac{VD\rho}{\mu}$$

When R_e is less than 2,100 the flow is Streamline.

When R_e is more than 10,000 the flow is Turbulent.

Between 2,100 and 10,000 the flow may be Transitional or it may be Turbulent.

Above about 4,000, water can be assumed to be in Turbulent flow.

Viscous liquids should not be assumed to be in Turbulent flow until the Reynolds number exceeds 10,000.

When the flow is Streamline the heat transfer rate is proportional to the cube root of the velocity.

When the flow is Turbulent the heat transfer rate is proportional to the eight-tenths power of the velocity.

At Transitional flows all kinds of queer relations have been reported and it is not possible to be precise. (See McAdams, Section 805.)

The viscosity of water is given in Section 797, Table LXXIX.

Suppose we are heating water from 100° F. to 160° F. in a calorifier with 2-in. tubes. The heat transfer rate of the particular calorifier is, say, 100 Btu/sq ft./° F./hr., and the velocity of flow is 1 ft./sec. We wish to know what improvement would result from putting up the flow to 2 ft./sec. and to 4 ft./sec.

The density of water between these temperatures changes little and can be taken at 62 lb./cu. ft. The viscosity from Table LXXIX is ·00043 lb./ft. sec. at 100° F. and ·000269 lb./ft. sec. at 160° F.

The Reynolds Number at 100° F. will be

$$\frac{VD\rho}{\mu} = \frac{1 \times \cdot 167 \times 62}{\cdot 00043} = 24{,}079$$

The flow is therefore turbulent, and the H.T.R. will vary as $V^{0 \cdot 8}$.

Log 1 = 0	Log 2 = ·301	Log 4 = ·602
0 × ·8 = 0	·301 × ·8 = ·2408	·602 × ·8 = ·4816
Antilog 0 = 1	Antilog ·2408 = 1·74	Antilog ·4816 = 3·03

Increase of heat transfer rate due to increase of velocity from 1 to 2 ft./sec. will be 1·74 times.

Increase of H.T.R. due to increase of velocity from 2 to 4 ft./sec. will be $\frac{3 \cdot 03}{1 \cdot 74} = 1 \cdot 74$ times.

Table XLVIII gives a few heat transfer rate-velocity relations starting from 100 Btu/sq. ft./° F./hr. with turbulent flow.

TABLE XLVIII. EFFECT OF WATER VELOCITY ON HEAT TRANSFER RATE

(*Turbulent Flow*)

RELATIVE LIQUID VELOCITY	HEAT TRANSFER RATE
1	100
2	174
3	241
4	303
5	362
6	419
7	474
8	527
9	580
10	630

Now suppose we wished to heat a 70 per cent. sugar solution over the same temperature range in the same calorifier. The viscosity at 100° F. will be about ·1 lb./ft. sec., and the density about 83 lb./cu. ft. so that :—

$$R_e = \frac{1 \times \cdot 167 \times 83}{\cdot 1} = 138 \cdot 6.$$

At 160° F. the viscosity will be about ·015 lb./ft. sec. This is a great reduction but not nearly enough to bring the Reynolds Number above the Streamline flow value. So the H.T.R. will vary as $V^{0 \cdot 33}$.

Log 1 = 0	Log 2 = ·301	Log 4 = ·602
0 × ·33 = 0	·301 × ·33 = ·0993	·602 × ·33 = ·1987
Antilog 0 = 1	Antilog ·0993 = 1·257	Antilog ·1987 = 1·580

Doubling the velocity in the case of this viscous liquid only increases the H.T.R. by 1·257 times ; $(\frac{1 \cdot 580}{1 \cdot 257} = 1 \cdot 257)$.

The increased velocity shows a marked improvement but, of course, the resistance of a multi-pass heater must be overcome by pumping. It is impossible to say what the resistance of a multi-pass heater will be. The calorifier makers have data based on experience. We have to balance the claims of a small heater, quick liquid passage and large pumping cost against little or no pumping cost, slow liquid passage and large calorifier.

When in doubt, a liquid velocity of about 4 ft./sec. is a good compromise (provided the liquid is not particularly viscous) between pumping power, calorifier size and liquid movement. Very viscous materials such as tar or molasses can only safely be dealt with on the basis of past experience.

335. VALVELESS PUMP. For effecting movement of viscous liquids in calorifiers or heat exchangers it is quite impossible to use multi-pass heaters because the power required would be out of all reason. It is possible to use a single pass heater and yet get quite a fast liquid movement if a slow-speed reciprocating valveless pump which uses quite a moderate amount of power, is put in the pipe line.

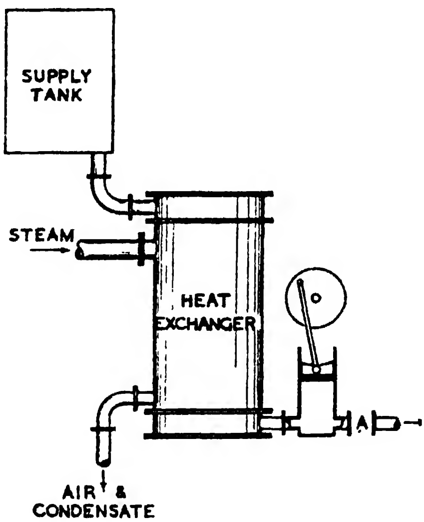


FIG. 170. SINGLE ACTING VALVELESS PUMP IMPARTING RECIPROCATING MOVEMENT TO MATERIAL IN HEAT EXCHANGER CLOSE TO THE SUPPLY TANK

Fig. 170 shows a valveless pump connected to the line in a system where the calorifier or heat exchanger is very close to the supply tank. The pump imposes a reciprocating movement on to the flow of liquid so that it flows say 2 ft. 3 in. forwards and 2 ft. back. Provided the discharge pipe is long or is controlled by a stop valve there is no need for any valve in the pump, in fact any valve in the pump would spoil the whole thing. The liquid surges to and fro through the heater in and out of the tank. If the discharge pipe is short or

open-ended it may be necessary to fit a constriction, by valve or orifice, at point A to prevent the pump reciprocating the liquid in the discharge pipe instead of through the heat exchanger.

Where the heater is some way from the supply tank, or where there is no supply tank to absorb the surge, the pump should be double acting, as shown in Fig. 171. The pump must still be valveless. One side of the piston absorbs the surge produced by the other side. If the supply and discharge lines are long and not too large they will accept none of the surge. This arrangement has great benefits. The lower the resistance offered by the calorifier the easier can the pump produce a good movement and the less power will be needed. The lower the resistance of the heater the less chance there is of the surge being transmitted to the supply or discharge pipes.

The valveless pump is not merely useful for viscous liquids. Wherever there is a low velocity heat exchanger whose heat transmission requires improving the valveless pump may be the economical answer.

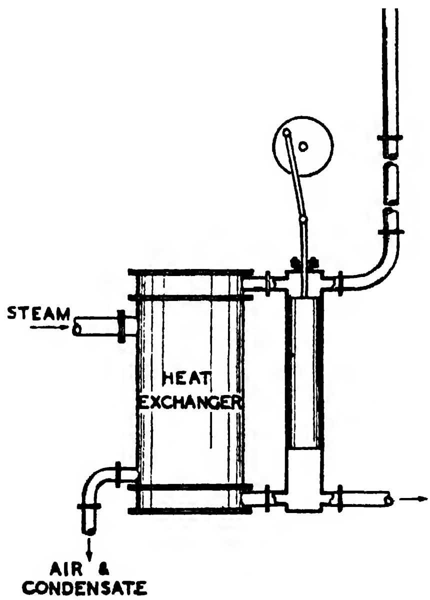
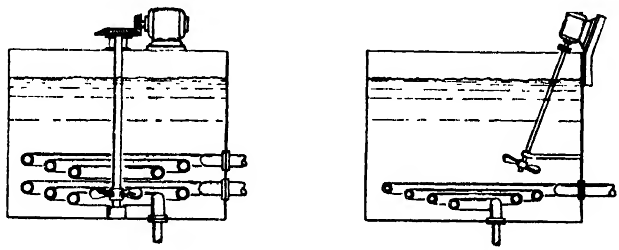


FIG. 171. DOUBLE ACTING VALVELESS PUMP IMPARTING RECIPROCATING MOVEMENT TO MATERIAL IN HEAT EXCHANGER IN THE MIDDLE OF A LONG PIPE LINE

336. **MOVEMENT IN TANKS AND VATS.** The natural circulation due to convection currents in a tank, vat or beck depends on the shape of the vessel, the arrangement and temperature of the heating surface and the properties of the liquid. It is almost impossible to lay down any general figure that would be trustworthy. One thing only is fairly certain, namely that natural circulation is usually totally inadequate for even modest heat transfer.

Movement can be improved by propellers, paddles or by pumping ; it is sometimes done by compressed air or open steam ; good results may be got by means of a displacer. Heat transfer in vats and tanks is often so bad that it deserves care and attention which will be well repaid. Many products suffer from local stewing in a heated tank ; often the coils get coated with a thick scale of burnt product, still further retarding heat transfer.

337. PROPELLER CIRCULATION. The propeller is tempting. It is cheap and small and easily fitted. The results are often disappointing. Arrangements such as are shown in Fig. 172 are not always satisfactory. The propeller is often too small, generally runs too fast and has the wrong shape of blade. To get good results the propeller should be shrouded in a sleeve, but this is difficult to arrange if the level of liquid in the tank varies. The arrangement shown in Fig. 172b is often little more than a gesture. Comparing power used with result obtained, propeller circulation is probably the least satisfactory of all the ways of mechanically moving tank contents.



a b

FIG. 172. PROPELLER CIRCULATORS

338. PADDLE CIRCULATION. There is much to be said for the good old five-barred gate trundling slowly round. It uses very little power. It is low-brow and simple. It is robust and cheap. The power needed is not great and there is little or no churning. (Much of the power put into propellers is wasted in local churning.) The liquid is made to pass fairly rapidly along the heating coils, but there may be a tendency for it to rotate in layers of different temperature and different density. These layers are to some extent broken up by the natural convection currents which can be augmented by giving the paddle bars a twist so that the liquid tends to be displaced downwards in the centre of the tank and upwards round the outside. Fig. 173 shows such a simple three-barred paddle with twisted arms.

339. PUMP CIRCULATION. Pump circulation is positive, but to ensure complete circulation of the whole of the tank contents means drawing from the bottom of the tank and discharging high up. If the tank is only quarter or half full this causes aeration, which may encourage oxidization, which may be inadmissible, or may cause foaming. For these reasons pump circulation is not always the easy method that it appears at first sight to be. It is very difficult with pump circulation, to make the forced circulation follow the natural circulation. Generally pump circulation is not a very good method, though there may be some situations where it is adequate and certain.

340. HIGH-CAPACITY LOW HEAD IMPELLERS. When the liquid in the vat or tank is always at a constant definite level a special form of high capacity low head impeller can often be used with great effect. The application is particularly useful where a very violent circulation is necessary, but the liquid level must be within a few inches of exactitude for the impeller to work properly.

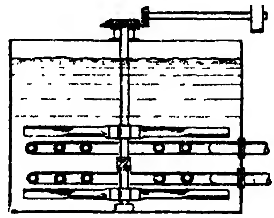


FIG. 173. PADDLE CIRCULATOR

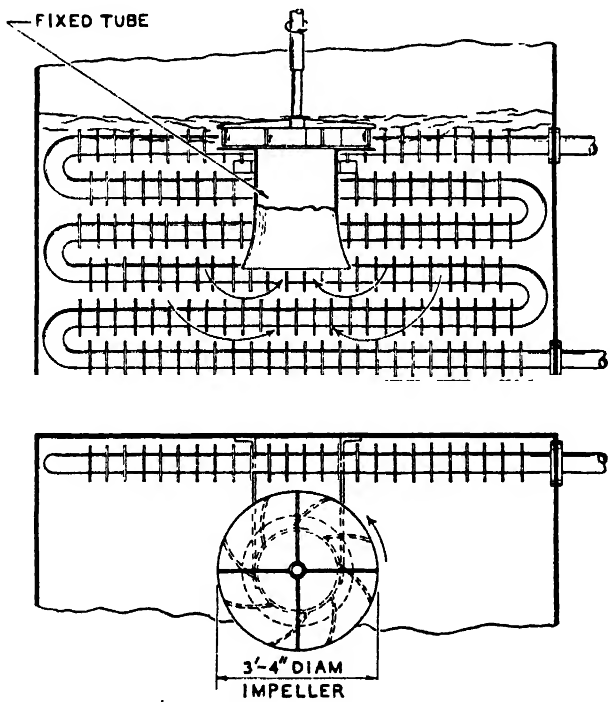


FIG. 174. HIGH-CAPACITY LOW HEAD CIRCULATING IMPELLER

Fig. 174 shows such an impeller in a tank in the author's factory, where the problem was to cool a sugar solution rapidly by means of coils containing refrigerated brine. The impeller is a large-diameter crude vaned rotor. It is just submerged and no more. The static head against it is therefore only its own depth. Being shrouded, it does not churn and is fairly efficient. It draws

the liquid from below the centre of the impeller and spreads it over the whole surface of the tank. The impeller shown in Fig. 174 circulates 6,500 gallons per minute for a power consumption of 5 H.P. at 70 r.p.m.

341. DISPLACER CIRCULATION. If a body with a large volume is placed in the tank and then moved, the liquid must flow into the space just vacated by the displacer. A displacer is shown fitted in a tank in Fig. 175 and its action is self-evident. As the displacer rises and falls the liquid flows to and fro across the coils. If the displacer is made about the same weight as the liquid it displaces and if it is roughly streamlined, it uses relatively little power. It is probably the most mechanically efficient of all forms of circulator, but the material in the corners remains stagnant, unless the corners are rounded to a big radius.

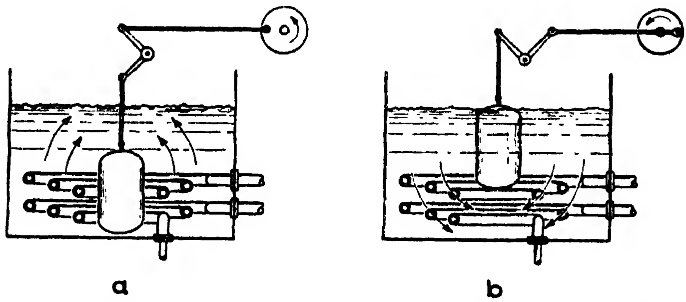


FIG. 175. SINGLE-ACTING DISPLACER FOR SMALL OR CIRCULAR VATS

Fig. 176 shows the method of applying the displacer principle to a long tank. A diaphragm plate is fitted in the middle of the upper part of the tank to compel the material to flow over the heating surface.

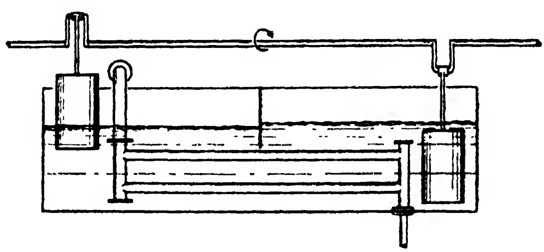


FIG. 176. DOUBLE-ACTING DISPLACER FOR LONG TANKS

342. AIR JET CIRCULATION. Circulation by means of air blown into the tank is often used. From a purely thermodynamical point of view this method has nothing to recommend it. Compressed air on expansion gets cold and takes good heat out of the product into which we are trying to put heat. Compressed air is about the most inefficient method of transmitting mechanical action. In some processes the oxidizing effect of air is detrimental ; air may also cause foaming. On the other hand there are some processes where an air stream blown through the liquid removes undesirable volatiles and where

some oxidization is wanted. In such cases air can be a really admirable circulating tool. The air jets take up little or no space ; they can be fitted in the most suitable part of the vessel. Violent agitation can be secured. If the air is used primarily for oxidization or for carrying away volatiles the air jets should probably be small. Perforations in a pipe may be adequate. If, however, the air is primarily to induce movement the air jets should be large—we want great footballs of air to act as displacers.

For this purpose a slow speed reciprocating air pump without an air receiver will give a good pulsating movement. By fitting a tube in the tank with the air jet at the bottom, we can get a positive air lift circulation up the tube and down the outside.

343. STEAM JET CIRCULATION. Steam jets for aiding circulation are seldom used in conjunction with heating surfaces save in some evaporators. If directly injected steam is permissible for heating there is no need to go to the expense of heating surface. The real criminals in the use of steam jets for circulation are some of the textile plants. Dye becks are brought up to heat by means of open steam and the steam is then used to promote circulation. This fills the dye-house with steam ; wastes much of the heat in the steam and has little to recommend it except technical tradition. It is a highly technical matter and must be left to the experts, but on the face of it live steam, for circulation only, is a most extravagant way of doing it and some dye-houses do very well without it.

344. FINS OR GILLS. Gilled pipes are not used nearly as much as they deserve to be. Where heat transfer resistance is principally made up of the heated film, as in the heating of air or of viscous liquids, a plain pipe is just silly. It has a relatively huge steam volume and a relatively low heat transfer. Steam flow is therefore very slow, which encourages the formation of air and condensate films inside the pipe. If plain pipes are placed very close together in order that viscous liquid shall not flow disinterestedly through the heater, a great obstacle is offered to the flow. Gills or fins slice the heated product up into rashers while offering very little resistance to flow. Fins or gills enable a large heating surface to be produced from a very small pipe, thus allowing brisk movement of the heating steam or water. There is of course no application for gills if the material is such as will clog the spaces between the fins or gills. It should be evident that with very viscous materials in sluggish flow it is necessary to provide a much greater heating surface outside the pipe to transfer the heat from pipe to product than is required inside the pipe to transfer the heat from the heating medium to the pipe.

Fig. 177a shows an arrangement of finned heating pipes applied to an exceedingly difficult heating problem in the author's factory. The " liquid " consists of cool molasses with 40 per cent. of sugar crystals in suspension. Its consistency is similar to that of warm asphalte as it is applied to a city street. The process aims at reducing the viscosity of the molasses by heat prior to separating the crystals. It is essential that the supersaturation of the molasses be not reduced to below saturation, or the crystals, so laboriously extracted, would be redissolved. The temperature of the heating medium—in this case water—must not therefore exceed the saturation temperature of the molasses.

The maximum permissible temperature difference is 45° F. The heat transfer rate is quite remarkable—3 Btu/sq. ft. of fins/hour/° F. diff. or the equivalent of 75 Btu/etc. from the plain surface of the pipe. The heater is a 3 foot pipe 35 feet long and the rate of flow is extremely small—4 inches per minute.

The fins are interrupted every two feet and their angular position turns slightly so as to prevent material lying stagnant in the upper V's.

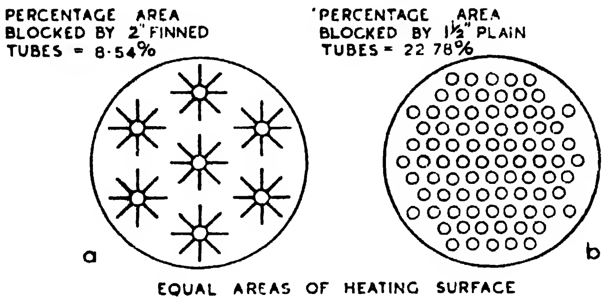


FIG. 177. FINNED HEATING SURFACE FOR ULTRA VISCOUS LIQUIDS AND EQUIVALENT HEATING SURFACE WITH PLAIN TUBES

Fig. 177b shows the cross section of the heater if plain pipes were used. The heating surface offers so much obstruction that there would be little or no flow of material. The obstructed cross section would be 23 per cent. of the total cross section, whereas with the finned pipes in Fig. 177a the heating surface only represents an obstruction of 8½ per cent.

In Fig. 174 the brine cooled pipes are provided with gills for a similar reason. Were plain pipes used the obstruction to circulation would be excessive. The rate of "cool" transfer in this process is 45 Btu/etc. equivalent to 102 Btu/etc. for the ungilled surface. The gills increased the surface by rather more than 100 per cent., and increased the Heat Transfer by 36 per cent.

345. WHAT IS THE HEATING SURFACE ? In a small diametered tube or in a thick-walled tube the inside surface of the tube is distinctly smaller than the outside surface. Which surface should be taken when trying to decide upon the heating surface that a particular job requires, or when trying to estimate the performance of a plant ?

The answer is really quite straightforward. In the books on heat transmission the matter is gone into at considerable mathematical length. In view of the fact that we do not know very much about heat transfer rates for different materials and plants there is no need for anything but the simplest of approximations. When the heating and the heated media are under approximately the same conditions and have similar properties, we can take the average of the inside and outside heating surfaces. When the conditions are very different we must take the heating surface that is doing the most difficult job.

Suppose we have an economiser with cast iron tubes 4 in. internal diameter and ⅝ in. thick. The inside surface will have an area of 1·047 sq. ft./ft. length

and the outside surface will be 1·44 sq. ft./ft. There is no doubt as to which surface to take. It is easy for the metal to part with its heat to the water but very difficult for the metal to take up heat from the gases. So the outside surface must be taken.

A copper tube ½ in. diameter and 16 S.W.G. will have an inside diameter of ·5 in. and an outside diameter of ·628 in. The inside surface will correspond to 7·64 feet of length to the sq. ft. and the outer surface will be 6·08 feet run to the sq. ft. If a water-cooled oil-cooler is made of ½ in. copper tube with the oil inside the tube, then we take the inside surface. If the oil is outside the tube then we take the outside surface because the oil is viscous and heat transfer is much lower between metal and oil than between metal and water.

Table XLIX shows the surface area of small bore pipes as linear feet per square foot of surface, and Table L shows the surface area of larger pipes as square feet of surface per foot run of pipe.

TABLE XLIX. FEET LENGTH PER SQUARE FOOT OF SURFACE OF TUBES OF SMALL DIAMETER

DIAMETER	·00″	·01″	·02″	·03″	·04″	·05″	·06″	·07″	·08″	·09″
0″		382	191	127	95·5	76·4	63·7	54·6	47·7	42·4
·1″	38·2	34·7	31·8	29·4	27·3	25·5	23·9	22·5	21·2	20·1
·2″	19·10	18·19	17·36	16·61	15·92	15·28	14·69	14·15	13·64	13·17
·3″	12·73	12·32	11·94	11·57	11·23	10·91	10·61	10·32	10·05	9·79
·4″	9·55	9·32	9·09	8·88	8·68	9·49	8·30	8·13	7·96	7·80
·5″	7·64	7·49	7·35	7·21	7·07	6·94	6·82	6·70	6·59	6·47
·6″	6·37	6·26	6·16	6·06	5·97	5·88	5·79	5·70	5·62	5·54
·7″	5·46	5·38	5·31	5·23	5·16	5·09	5·02	4·96	4·90	4·84
·8″	4·77	4·72	4·66	4·60	4·55	4·49	4·44	4·39	4·34	4·29
·9″	4·24	4·20	4·15	4·11	4·06	4·02	3·98	3·94	3·90	3·86
1·0″	3·82	3·78	3·74	3·71	3·67	3·64	3·60	3·57	3·54	3·50
1·1″	3·47	3·44	3·41	3·38	3·35	3·32	3·29	3·26	3·24	3·21
1·2″	3·18	3·16	3·13	3·11	3·08	3·06	3·03	3·01	2·98	2·96
1·3″	2·94	2·92	2·89	2·87	2·85	2·83	2·82	2·80	2·78	2·76
1·4″	2·73	2·71	2·69	2·67	2·65	2·63	2·62	2·60	2·58	2·56
1·5″	2·55	2·53	2·51	2·50	2·48	2·46	2·45	2·43	2·42	2·40
1·6″	2·39	2·37	2·36	2·34	2·33	2·31	2·30	2·29	2·27	2·26
1·7″	2·25	2·23	2·22	2·21	2·20	2·18	2·17	2·16	2·15	2·13
1·8″	2·12	2·11	2·10	2·09	2·08	2·06	2·05	2·04	2·03	2·02
1·9″	2·01	2·00	1·99	1·98	1·97	1·96	1·95	1·94	1·93	1·92
2·0″	1·91	1·90	1·89	1·88	1·87	1·86	1·85	1·85	1·84	1·83
2·1″	1·82	1·81	1·80	1·79	1·78	1·78	1·77	1·76	1·75	1·74
2·2″	1·74	1·73	1·72	1·71	1·71	1·70	1·69	1·68	1·68	1·67
2·3″	1·66	1·65	1·64	1·64	1·63	1·63	1·62	1·61	1·60	1·60
2·4″	1·59	1·58	1·58	1·57	1·57	1·56	1·55	1·55	1·54	1 53
2·5″	1·53	1·52	1·52	1·51	1·50	1·50	1·49	1·49	1·48	1·47
2·6″	1·47	1·46	1·46	1·45	1·45	1·44	1·44	1·43	1·43	1·42
2·7″	1·41	1·41	1·40	1·40	1·39	1·39	1·38	1·38	1·37	1·37
2·8″	1·36	1·36	1·35	1·35	1·34	1·34	1·34	1·33	1·33	1·32
2·9″	1·32	1·31	1·31	1·30	1·30	1·29	1·29	1·29	1·28	1·28

TABLE L. SQUARE FEET OF SURFACE PER FOOT OF LENGTH OF TUBES OF LARGE DIAMETER

DIAMETER	·0″	·1″	·2″	·3″	·4″	·5″	·6″	·7″	·8″	·9″
3·0″	·787	·811	·838	·864	·890	·916	·942	·969	·995	1·021
4·0″	1·047	1·073	1·100	1·126	1·152	1·178	1·204	1·230	1·257	1·283
5·0″	1·309	1·335	1·361	1·388	1·414	1·440	1·466	1·492	1·518	1·545
6·0″	1·570	1·597	1·623	1·649	1·676	1·702	1·728	1·754	1·780	1·806
7·0″	1·833	1·859	1·885	1·911	1·937	1·963	1·990	2·016	2·042	2·068
8·0″	2·09	2·12	2·15	2·17	2·20	2·23	2·25	2·28	2·30	2·33
9·0″	2·36	2·38	2·41	2·43	2·46	2·49	2·51	2·54	2·57	2·59
10·0″	2·62	2·64	2·67	2·70	2·72	2·75	2·78	2·80	2·83	2·85
11·0″	2·88	2·91	2·93	2·96	2·98	3·01	3·04	3·06	3·09	3·12
12·0″	3·14	3·17	3·19	3·22	3·25	3·27	3·30	3·32	3·35	3·38
13·0″	3·40	3·43	3·46	3·48	3·51	3·53	3·56	3·59	3·61	3·64
14·0″	3·67	3·69	3·72	3·74	3·77	3·80	3·82	3·85	3·87	3·90
15·0″	3·93	3·95	3·98	4·01	4·03	4·06	4·08	4·11	4·14	4·16
16·0″	4·19	4·21	4·24	4·27	4·29	4·32	4·35	4·37	4·40	4·42
17·0″	4·45	4·48	4·50	4·53	4·56	4·58	4·61	4·63	4·66	4·69
18·0″	4·71	4·74	4·76	4·79	4·82	4·84	4·87	4·90	4·92	4·95
19·0″	4·97	5·00	5·03	5·05	5·08	5·11	5·13	5·16	5·18	5·21
20·0″	5·24	5·26	5·29	5·31	5·34	5·37	5·39	5·42	5·45	5·47
21·0″	5·50	5·52	5·55	5·58	5·60	5·63	5·65	5·68	5·71	5·73
22·0″	5·76	5·79	5·81	5·84	5·86	5·89	5·92	5·94	5·97	6·00
23·0″	6·02	6·05	6·07	6·10	6·13	6·15	6·18	6·20	6·23	6·26
24·0″	6·28	6·31	6·34	6·36	6·39	6·41	6·44	6·47	6·49	6·52

346. HEATING TRANSMISSION PIPES. Where viscous fluids must be piped long distances great resistance is offered to flow unless the size of the pipe is beyond reason. Where the pipe is in fairly constant use it should be jacketed and warm water, possibly waste condenser water, can be passed through the jacket. Jacketed pipes can be purchased and are catalogued. Where the pipe is only in occasional use the most practical way is to wind a helix of insulated wire round the pipe between the pipe surface and the lagging and pass current through. Quite a modest current will work wonders. In the author's factory a molasses delivery pipe 180 ft. long and 12 inches diameter with a fall of 1 in 23, is used once or twice a week. The time of reaching full flow has been reduced by between 15 and 45 minutes dependent on the weather by winding a coil of wire round the pipe. The current consumption is only 2¾ kW.

347. NO HEAT TRANSFER. Many years ago a product made its appearance in the author's factory which appeared in no way different from the normal product produced at this part of the process. This product requires concentration and crystallisation so that the starting water content of about 40 per cent. is reduced to about 7 per cent. The process normally takes about 7 hours. With this material no appreciable heat transfer took place in 14 hours. The material was examined for heat conductivity, viscosity, specific heat, etc., but no physical characteristic appeared to be in any way abnormal, nor was there anything the matter with the plant or the heating steam. The material

(which had its origin in Poland) just refused to boil. The mystery was never solved and the material had to be mixed away with normal material.

348. DROP CONDENSATION. The surface of a liquid behaves rather like a very thin elastic bag. The effect of this "surface tension" is seen in many everyday sights. A drop of water immediately tries to become a sphere. The tension of the surface tries to make the surface as small as possible—just like a rubber balloon. The water in a glass will stand quite high above the rim of the glass until the hydrostatic head of the water overcomes the surface tension. A drop of water dropped into oil or petrol will not spread as a thin layer over the bottom of the container. The water gathers itself together into a ball, or if it is large, into a bun. Water has a very high surface tension and is always trying to gather itself into drops. When water is deposited on a surface it tries to gather itself together into drops. It will do this on oily glass. Polished nickel particularly objects to being "wetted" by water. A rough dirty surface "wets" easily. The reasons that govern the wetting of surfaces are imperfectly understood—at any rate by the author. If a heating surface could be made in such a way as not to wet, heat transfer would be very greatly improved because immediately a film of water a few molecules thick had deposited by condensation it would gather itself together into a drop. This is the principle of "Drop Condensation" about which a good deal has been written in recent years. For most ordinary purposes there is no need to consider such a refinement. But, if for some reason the plant must be reduced at all costs to the smallest possible size, it might be worth while trying to ensure drop condensation inside a steam heating surface. Unless all the other resistant films can be reduced to almost negligible effects there is seldom any marked gain to be obtained from trying to eliminate the condensate film. There are also certain practical difficulties. Suppose the heating surface can be made to induce drop condensation, how can one be sure that it is going to remain so? The least film of oxide or scale may upset the whole business. There are very few plants in which the inside of a heating surface can be examined. In certain types of jacketed pan the heating surface is made to be accessible.

Accounts of a certain amount of work on drop condensation have appeared during the last few years in technical journals. The following points appear to have been established :—

1. Perfectly clean steam always condenses filmwise on a perfectly clean surface.
2. After a time the slight oily impurities in many industrial steams are sufficient to alter a perfectly clean surface so that drop condensation commences. In fact with some steam it was found impossible to prevent drop condensation taking place on certain clean surfaces.
3. Drop condensation occurs more readily on smooth than on rough surfaces.
4. Drop condensation cannot be maintained on mild steel or aluminium surfaces.
5. Drop condensation can be induced, if the steam does not contain the necessary impurities, by the injection of minute traces of fatty acids, e.g. oleic acid, which is particularly successful on brass, nickel, chromium or copper. Other substances, e.g. mercaptans, can be used as drop promoters.
6. Some drop promoters are only effective where there is a trace of non-condensible gas in the steam.

Now it is clear that in an evaporator, while we want to induce and maintain drop condensation on the steam side so that the surface is wetted as little as possible, we want exactly the opposite on the liquid side. Every possible means should be taken to ensure that the liquid side is " wetted " as much as possible and that " film boiling " does not occur and interpose a layer of steam between the surface and the liquid. The boiling should be in the form of minute bubbles which quickly detach themselves—this is called " nucleate boiling ".

349. LONG-PIPE HEATERS. Heating by a single long pipe has two main applications ; the space heating of many of the older textile mills, and the heating of oil tanks ashore and afloat. Long pipe heating may use either hot water or steam. Each presents its own unsatisfactory problems.

Consider first hot water. At the beginning of the long pipe the water is hot and heat transfer is good. Further along the pipe the water is cooler, the temperature drop is lower, the viscosity of the water is higher so that the heat transfer rate drops still more. If the pipe is made progressively smaller in bore, the water velocity will increase as the water gets cooler. This is one way of improving matters, but it can hardly be used without forced circulation. The better way is to circulate the water so rapidly that there is very little temperature drop over the whole length.

Now consider steam. Steam is condensing all along the pipe ; condensate is accumulating ; there is a continuous pressure drop all along the pipe. At the beginning of the pipe the steam pressure is highest, there is little condensate and little air so that heat transfer is good. At the end of the pipe there is little pressure, the pipe is probably half full of condensate with the other half full of air. Steam heated long pipe heaters are not satisfactory. There are two possible aids. The steam can possibly be circulated (see Chapter 18) or the long pipe can be split up into smaller lengths—in other words abolish the long pipe.

If long pipe heating must be done high velocity water—pressure hot water if necessary, see Chapter 18—is probably the best solution, but the pumping power may be high. If steam must be used the pipe should be as small in bore as possible so as to raise the ratio of heating surface to pipe volume and thus increase the flow of steam in the pipe. The pipe must have a really good fall and there must be proper air vents. The pressure must be fairly high.

350. MEAN TEMPERATURE DIFFERENCE. In all types of heater where the temperatures on one or both sides of the heating surface change, the temperature difference is not a constant figure and we must have some way of arriving at the real mean temperature difference. There are quite a number of possible variations and we must consider them briefly.

Let us first look at a simple batch heating job in a tank or vat, heated by steam and properly trapped and vented. The temperature conditions are shown in Fig. 178. The steam temperature is constant, the temperature of the material in the vessel can be assumed to be more or less uniform but is rising with time. At first, when the temperature difference is great, the liquid will heat up quickly. The hotter it gets the less will be the temperature difference and the slower will the temperature increase. The mean temperature difference

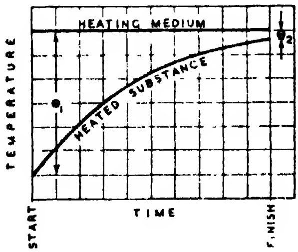


FIG. 178. TEMPERATURE DIFFERENCE WHEN HEATING TANK OR VAT BY STEAM

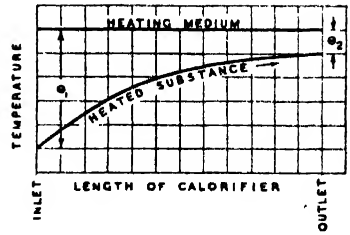


FIG. 179 TEMPERATURE DIFFERENCE WHEN HEATING LIQUID IN CALORIFIER BY STEAM

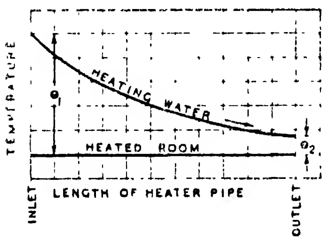


FIG. 180. TEMPERATURE DIFFERENCE WHEN HEATING A ROOM BY WATER IN A LONG PIPE HEATER

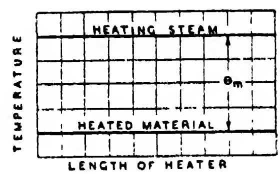


FIG. 181. TEMPERATURE DIFFERENCE IN AN EVAPORATOR OR SINGLE ROW AIR HEATER STEAM HEATED

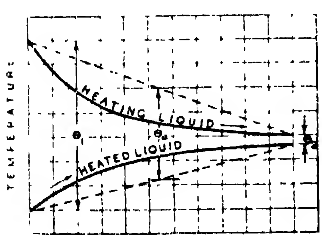


FIG. 182. TEMPERATURE DIFFERENCE IN A CON-CURRENT HEAT EXCHANGER

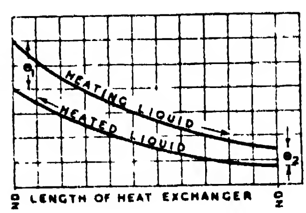


FIG. 183. TEMPERATURE DIFFERENCE IN A COUNTERCURRENT HEAT EXCHANGER

could be found by taking temperatures every minute and averaging the result, which, had the time intervals been constant and sufficiently frequent, would be the true mean temperature difference.

Now consider a calorifier. Here there is a flow, but time does not come into the picture, the operation is continuous and is independant of time. Fig. 179 shows a steam heated calorifier into which a cool liquid is flowing and being heated. The steam temperature is constant but the liquid temperature rises along the length of the calorifier. It might be possible to take temperatures at regular equal intervals along the calorifier and thus obtain the true average temperature difference. The temperatures all along the calorifier will vary with varying rates of flow and such a reading of multiple temperatures would generally be impractical.

Now consider a long-pipe room heater using hot water. Fig. 180 shows the conditions. The room temperature will be virtually constant, but the water temperature will fall along the length of the pipe. This fall in water temperature will be rapid at first and will tail off as the temperature of the water drops. The temperature at even distances along the pipe could be measured to find the true mean temperature difference.

Fig. 181 shows the conditions where there is no change of temperature with either time or position. This would be the state of affairs in an evaporator or in an air heater consisting of a single row of steam heated tubes. There is here only one temperature difference and hence no problem.

Fig. 182 shows a con-current heat exchanger, and Fig. 183 a countercurrent heat exchanger. Here both liquids are changing temperature and it would be practically impossible to take accurate evenly spaced temperatures in both liquid passes, especially if the exchangers were multi-pass.

Owing to the curvature of the temperature/time or temperature/distance lines the arithmetical mean of the two temperature differences, at the beginning and end, would be clearly wrong. The possible error is clearly shown in Fig. 182 by θ_a.

Fortunately it has been found that a single relation between the initial and final temperature differences will give the true mean temperature difference in all of these curved temperature relations whether in respect of time or distance. The formula is :—

$$\theta_m = \frac{\theta_i - \theta_f}{2\cdot 3 \log \dfrac{\theta_i}{\theta_f}} \quad \text{or} \quad \left(\frac{\theta_i - \theta_f}{\log_e \dfrac{\theta_i}{\theta_f}}\right)$$

Where θ_m = Mean Temperature Difference wanted.
θ_i = Initial Temperature Difference.
θ_f = Final Temperature Difference.

Hausbrand (Section 805) gives what he suggests is a simpler formula (blessed by Badger) to replace that given above on the grounds that a formula containing logarithms is inconvenient. But the simpler formula necessitates reference to a table, whereas logs can be taken out, with sufficient accuracy for the present purpose, from a pocket slide rule.

(If a table of natural logarithms is available the calculation can be slightly shortened by using the formula in brackets.)

Example 1.

Fig. 178. Steam heated vat.

Steam temperature	230° F.
Initial liquid temperature	80° F.
Final liquid temperature	185° F.

Then $\theta_i = 230 - 80 = 150$

$\theta_f = 230 - 185 = 45$

$\dfrac{\theta_i}{\theta_f} = 3{\cdot}33 \qquad \log 3{\cdot}33 = {\cdot}522 \qquad 2{\cdot}3 \times {\cdot}522 = 1{\cdot}2$

$\theta_i - \theta_f = 150 - 45 = 105$

$\theta_m = \underline{\underline{87{\cdot}5}}$

The arithmetic mean is $\dfrac{150 + 45}{2} = 97{\cdot}5$. Error 11 per cent.

Example 2.

Fig. 180. Water-heated long pipe.

Room temperature	62° F.
Water inlet	165° F.
Water outlet	125° F.

$\theta_i = 165 - 62 = 103$

$\theta_f = 125 - 62 = 63$

$\dfrac{\theta_i}{\theta_f} = 1{\cdot}635 \qquad \log 1{\cdot}635 = {\cdot}2135 \qquad 2{\cdot}3 \times {\cdot}2135 = {\cdot}49$

$\theta_i - \theta_f = 40$

$\theta_m = \underline{\underline{81{\cdot}6}}$

Arithmetic mean $= \dfrac{103 + 63}{2} = 83$. Error 2 per cent.

Example 3.

Fig. 182. Concurrent heat exchanger.

Heating water inlet	198° F.
Heated water inlet	57° F.

$\theta_i = 198 - 57 = 141$

Heating water outlet	102° F.
Heated water outlet	91° F.

$\theta_f = 102 - 91 = 11$

$\dfrac{\theta_i}{\theta} = 12{\cdot}8 \qquad \log 12{\cdot}8 = 1{\cdot}107 \qquad 2{\cdot}3 \times 1{\cdot}107 = 2{\cdot}55$

$$\theta_i - \theta_f = 141 - 11 = 130$$

$$\theta_m = \underline{\underline{51}}$$

Arithmetic mean $= \dfrac{141 + 11}{2} = 76$. Error 49 per cent.

From these examples it will be seen that where the initial and final temperature differences are not greatly unlike the arithmetic mean temperature difference may well be good enough. But where the initial and final temperature differences are far apart the error introduced by using the arithmetic mean may be very large.

The errors introduced by taking the arithmetic mean temperature difference instead of the log mean temperature difference are :—

When θ_f is 70 per cent. or more of θ_i the error is less than 1 per cent.
When θ_f is 50 per cent. or more of θ_i the error is less than 4 per cent.
When θ_f is 33 per cent. or more of θ_i the error is less than 10 per cent.
When θ_f is 20 per cent. or less of θ_i the error is more than 20 per cent.
When θ_f is 10 per cent. or less of θ_i the error is more than 40 per cent.

These are proved thus :—

If θ_f is expressed as a percentage, p, of θ_i, then

$$\theta_f = \frac{p}{100}\,\theta_i$$

The log mean formula can be rewritten

$$\theta_m = \frac{\theta_i\left(1 - \dfrac{p}{100}\right)}{2 \cdot 3 \ \log \dfrac{100}{p}}$$

The arithmetic mean $\theta_a = \dfrac{\theta_i + \theta_f}{2}$

$$= \frac{\theta_i\left(1 + \dfrac{p}{100}\right)}{2}$$

If the functions $\dfrac{\left(1 - \dfrac{p}{100}\right)}{2 \cdot 3 \log \dfrac{100}{p}}$ and $\dfrac{\left(1 + \dfrac{p}{100}\right)}{2}$

are compared for various values of p we get the results shown above.

In Section 345 it was stated that when finding the heating surface of a pipe or tube, the arithmetic mean of the inside and outside diameters could be used, if the conditions on both sides of the heating surface were similar. It was also stated that in most problems one surface usually had a much lower heat transfer rate than the other, and that the surface to be used should always be that on which heat transfer was most difficult. For real accuracy when the heat

transfer rate on either side of the pipe is the same, the log mean heating surface should be used. The errors introduced by taking the arithmetic mean are exactly the same as those associated with arithmetic mean temperature difference. As there are seldom cases in which the inside diameter is one half or less than the outside diameter, the error introduced by using the arithmetic mean will not normally exceed 4 per cent.

351. CHLORINATION. In calorifiers and surface condensers the water side of the heating surface often gets coated with slime, which offers considerable resistance to heat transfer. One of the best methods of preventing sliming is by adding doses of chlorine to the water. Of course the use to which the water is to be put must allow of the addition of chlorine, but the chlorine dose can be very small. Much better results are obtained by adding the chlorine in intermittent fairly strong doses than by continuous addition. All water treatment firms can supply details of chlorinators and give advice about chlorination.

* * *

The subject of drying--apart from the supply of steam heat to the heating pipes—is not dealt with in this book. It would require much space and is a subject, in itself, big enough to fill a book. As far as heat transfer from steam or hot water to air is concerned there are three simple rules for securing optimum heat transfer.

Keep the surface that heats the air clean.
Maintain a good air velocity.
Have a generous metal to air surface—fins or gills.

* * *

Heat transfer in boiling pans, crystallising pans and evaporators is a somewhat specialised subject, so it will be given a chapter to itself, which those who are not interested can skip.

* * *

CHAPTER 12

HEATING IN PANS AND EVAPORATORS

> The vacuum pan . . . is a copper shell composed of two hemispheroids—the lower one embedded in a steam jacket, the upper one not so embedded. In addition to the steam jacket external to the lower hemispheroid, there is, internal to the same, and coiling spirally within, a copper steam tube technically called a worm.
>
> J. G. MCINTOSH. *Technology of Sugar.* 1916.

THERE are many types of evaporator and many types of pan. They may be used for cooking, for concentrating or for crystallising. They may range in size from 10 gallons capacity to 100 tons. They may use steam at anything from 250 psi to 20-in. vacuum. They have some things in common ; almost all of them rely on natural circulation ; the heat transfer rate is generally higher than in any other type of plant.

No detailed description of all the different types will be given here ; only enough to bring out certain points. This Chapter is lop-sided, as pans and evaporators are key plants in the sugar industry ; consequently the author has much more first-hand experience with sugar pans and evaporators than with any other kind.

352. TYPES OF EVAPORATING PLANT. Fig. 184 shows some of the many types of evaporating plant :—

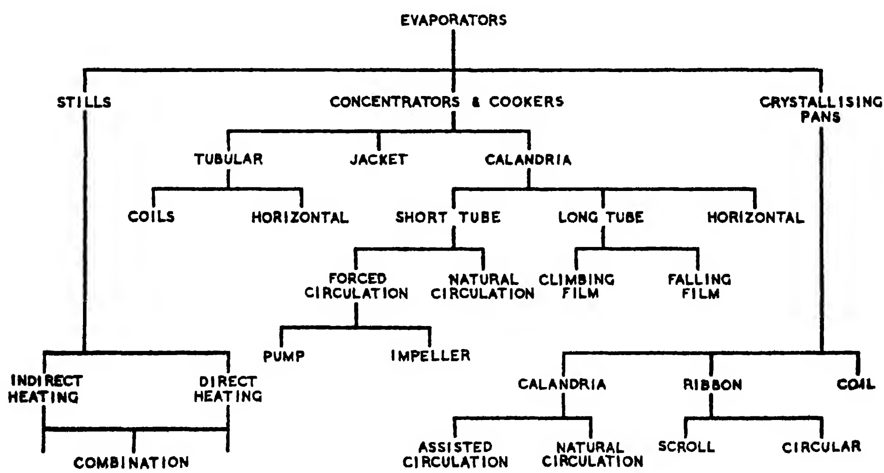


FIG. 184. EVAPORATOR TYPES

The heating-surface/material-volume ratio of a pan or evaporator, i.e, the sq. ft. of heating surface per cu. ft. of material capacity of the vessel, is a useful figure. It tells us roughly the "power" of the vessel. It indicates

the steam pressure that will be needed to get a given heat input. In a continuous evaporator the ratio will give us an idea of how long the material will be in process. Table LI shows the heating surface, the product volume and the surface/volume ratio of a number of vessels which will be discussed in certain particulars in later sections.

TABLE LI. SURFACE/VOLUME RATIOS OF SOME PANS AND EVAPORATORS

VESSEL		HEATING SURFACE SQ. FT.	MATERIAL VOLUME CU. FT.	SURFACE / VOLUME RATIO
Hemispherical jacketed boiling pans—				
1 ft. 6 in. diameter		7·07	2·22	3·18
2 ft. „		12·56	5·24	2·40
2 ft. 6 in. „		19·61	10·23	1·92
3 ft. „		28·28	17·67	1·60
4 ft. „		50·27	41·83	1·20
5 ft. „		78·53	81·82	·96
Coil crystallising pan..	(Fig. 191b)	554	718	·77
Coil crystallising pan..	(Fig. 192)	914	709	1·29
Crystallising pan—Coil	(Fig. 193)	2,263	1,200	1·89
Crystallising pan—Calandria	(Fig. 194)	2,191	1,200	1·83
Crystallising pan—Gräntzdörffer	(Fig. 195)	2,282	1,200	1·90
Crystallising pan—Scroll	(Fig. 196)	1,720	1,738	·99
Small standard Calandria evaporator	(Fig. 165)			
Calandria submerged	(Fig. 165)	1,436	225	6·4
Liquid ½ up tubes	(Fig. 165)	1,436	110	13
Horizontal film evaporator—Lillie ..	(Fig. 199)	296	12 ?	25 ?
Climbing film evaporator—Kestner	(Fig. 198)	104	4 ?	26 ?
Falling film evaporator—Kestner ..	(Fig. 198)	280	2 ?	140 ?

The surface/volume ratio taken alone can be misleading. It can only be used to compare like with like or for drawing distinctions. For example, the huge surface/volume ratios of the film evaporators give no idea of their heat transfer compared to a standard calandria evaporator. The real virtue lying behind their big surface/volume ratio is not exceptionally fast heat transfer, but the exceedingly short time the material being evaporated is in process. This property is very valuable when concentrating certain delicate organic products.

The temptation to increase heating surface in order to improve output is difficult to resist, but the extra obstruction to circulation often more than counteracts the value of the increased surface.

353. **TEMPERATURE VERSUS SURFACE.** Suppose we wish to increase the output of an evaporator. Better results will probably be obtained by doubling the temperature difference than by doubling the heating surface. Fig. 168, Section 331 shows that as temperature difference is increased, heat transfer rate is also increased. Suppose a liquid is being boiled at 212° F. with

(87684)

a temperature difference of 20° F., the H.T.R. from Fig. 168 is 440, and the heat transfer will be $440 \times 20 = 8{,}800$ Btu/sq. ft./hr. If the temperature difference is doubled to 40° F., Fig. 168 gives the H.T.R. as 550, so that the heat transfer will be $550 \times 40 = 22{,}000$ Btu/sq. ft./hr.

Increase of heating surface should increase heat transfer proportionally. Up to a point this does happen, but increase of heating surface almost always entails increase of obstruction, and there comes a point where the circulation is so greatly hampered that any further increase in heating surface actually reduces the total heat transfer.

It is sometimes suggested that high temperature heating surfaces may cause burning of the product. Burning of the product can only occur if the product is stagnant. Quick heat transfer with unimpeded circulation gives the least opportunity for stagnation and consequently for burning.

High temperatures inside the heating surfaces call for high steam pressures which are not conducive to steam economy, but, while steam economy is the main purpose of this book, good process technique is of paramount importance in a process factory. Provided all the flash steam is recovered from the high pressure condensate, and provided there is no extra radiation loss, there is no thermal loss with high pressure steam compared to low pressure steam ; the only loss is in the potential loss of power generation. High pressure steam cannot have produced the same amount of power as could the low pressure exhaust steam from an engine.

354. HEMISPHERICAL JACKETED PANS. These small pans are used in great numbers in many industries, particularly in the food industries—jam-boiling, confectionery boiling, soup and sauce cooking, etc. Their steam-locking and air-locking propensities have been discussed in Sections 304 and 322. In small sizes, provided the air and condensate removal has been properly dealt with, they are ideal for their purpose. Table LI shows how the surface/volume ratio falls off as their size increases. Their simple construction enables them to be cleaned quickly, easily and completely—this is very important in many food applications. In some designs the jacket is bolted to the body in such a way that the jacket can be readily dropped. In some factories the jacket is dropped every few days and the body heating surface is polished and lightly greased in an endeavour to secure drop condensation—see Section 348. In some cases the product has to be stirred. The clean smooth interior makes this easy.

In the larger sizes, as the surface-volume ratio falls off, output drops and the advantage of the larger size is greatly discounted. In an effort to improve the performance, coils are often added. These impede circulation, make cleaning really difficult and offer obstructions to the stirring paddle.

The small jacketed pan is a beautiful little piece of plant, but in large sizes it is not so good.

355. EVAPORATION UNDER VACUUM. There are two reasons for boiling a product under vacuum. The first is that for a given steam pressure the temperature difference is larger ; this generally permits of faster heat

transfer, or enables a lower steam pressure to be used. The second is to boil at a low temperature a product that deteriorates at high temperature. Any plant that calls for vacuum boiling for the second reason automatically has any advantages associated with the first reason.

Vacuum vessels can be heated by jackets, coils or calandrias. They may be used as evaporators for concentrating liquids, or for distilling, or they may be primarily crystallising vessels. Whatever their purpose or their design, the one essential quality that they have in common is that the boiling point of the liquid is reduced.

Table LII shows the vacua and the corresponding boiling points of a 60 per cent. sugar solution, and the steam pressure that would be needed to give a constant temperature difference of 80° F. across the heating surface.

TABLE LII. VACUA AND BOILING POINTS OF 60 PER CENT. SUGAR SOLUTIONS AND STEAM PRESSURES NEEDED TO GIVE 80° F. DIFFERENCE

VACUUM	BOILING POINT	STEAM PRESSURE
IN.	°F.	PSI.G.
0	217	50
5	210	43
10	196	32
15	184	23
20	165	13
25	138	2

Actually this table paints the steam pressure reduction in much too rosy colours. The figures are true enough. These are the pressures to give the constant temperature difference, but at the lower boiling temperatures a greater temperature difference is needed to give the same total heat transfer. Fig. 168 shows the relation between boiling temperature and heat transfer rate, and also shows the relation between temperature difference and heat transfer rate. From a study of Fig. 168 it might well be thought that instead of reducing the steam pressure as the boiling point is reduced, it might even be necessary to increase it. Fig. 168 was drawn from data obtained by Badger in the laboratory with distilled water. The matter merits some discussion even if only to emphasise that heat transfer obtained in one evaporator on one material must not be used to foretell performance in another evaporator on a different material.

Badger implies that the principal cause for the falling off in heat transfer rate at low temperature is due to the increased viscosity of the material being

evaporated, in this case water. If we take the heat transfer rate of 350 from Fig. 168 and assume that this is the rate we require we can tabulate thus :—

Temperature	*Vacuum*	*Viscosity*	*Temperature difference*	*Required steam pressure*
212° F.	0	·284	12° F.	4 psi.g.
167° F.	18·5 in.	·380	23° F.	11 in. vac.
140° F.	24 in.	·469	47° F.	12 in.
122° F.	26·25 in.	·549	83° F. (extrapolated)	4 in.

If this is the performance on distilled water, what might be expected with a viscous material like concentrated sugar liquor, whose viscosity is 50 to 200 times that of water ? It would seem likely that as the vacuum on the liquor increased the steam pressure might also have to be increased. Here is a comparison of the viscosities of water and 72 per cent. sugar solutions :—

Temperature	*Viscosity—Centipoises* Water	*Viscosity—Centipoises* 72 per cent. sugar solution
212° F.	·284	11·0
167° F.	·380	28
140° F.	·469	58
122° F.	·549	100

Now sugar boiling technique depends upon accurate and quick control of boiling rate, or heat transfer, and this is done entirely by adjustment of the vacuum under which the solution is boiling ; the steam pressure is generally left constant. From a consideration of Badger's data and the viscosities given above an increase in vacuum with constant steam pressure should have little positive, but possibly a negative, effect on heat transfer. This is not the case. There is a great jungle of ignorance regarding heat transfer that badly needs exploring. It is important that heat transfer rate figures obtained on one plant and on one material should not be used to foretell the heat transfer rate on another plant and/or on another material.

Whatever the heat transfer rate, there are a number of materials that must be evaporated under vacuum if damage is not to be done to them—sugar solutions, milk, fruit juices, hormone extracts, etc.

357. EFFECT OF BOILING TEMPERATURE. There seems little doubt that with many liquids the actual boiling temperature has a much greater proportional effect on heat transfer rate than has temperature difference. Fig. 185 shows a relation plotted for certain sugar evaporators. From this and from Fig. 168 it is clear that vacuum boiling has a very detrimental effect on heat transfer rate and hence calls for much larger plant or for a bigger temperature difference. The whole art of evaporation, like many other technologies, is the reconciliation of conflictions. The arguments in favour of vacuum evaporation are that boiling can be done at really cool temperatures, without injury to certain products, and that multiple effect evaporation can be carried over a much wider range.

358. FORCED CIRCULATION EVAPORATORS. With good design natural circulation is generally sufficient, but there are certain situations where

forced circulation is definitely beneficial. If the material being concentrated is corrosive, so that the evaporator must be built of expensive metal, stainless steel or nickel, it may pay to increase the heat transfer rate greatly by a really rapid circulation of the liquid and so reduce the size and cost of the evaporator.

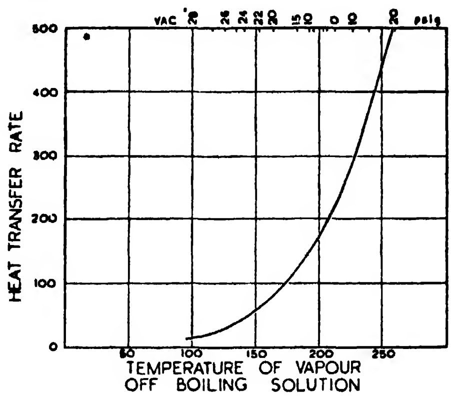


FIG. 185. EFFECT OF REDUCED BOILING PRESSURE ON HEAT TRANSFER RATE

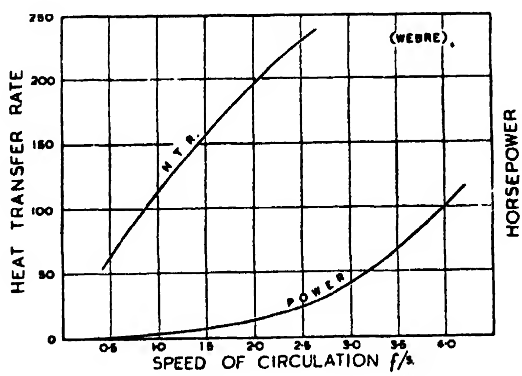


FIG. 186*. EFFECT OF ARTIFICIAL CIRCULATION ON HEAT TRANSFER RATE AND RELATIVE POWER NEEDED TO PRODUCE CIRCULATION

It is difficult to measure the rate of circulation in a working evaporator. Some measurements have been made using Pitot tubes. Webre gives results obtained on brine evaporation in the laboratory and these are shown in Fig. 186. The power curve given in Fig. 186 shows the power taken by a circulator in a very large working evaporator. The material being processed is not stated. This curve is also given by Badger. If an outside circulator is used the rate of circulation can be easily measured, but if the circulator is internal, measurement is very difficult.

* By permission of the Reinhold Publishing Company.

Another use for circulation is where scale is deposited on the heating surface. It has been found with certain scale-depositing solutions, for example common salt, that if the velocity over the heating surface is very high there is a greatly decreased scale deposit. This effect had also been noted with sugar

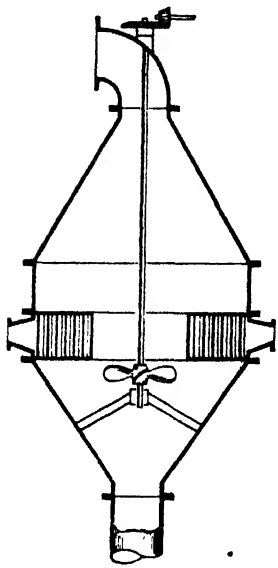


FIG. 187. ASSISTED CIRCULATION CALANDRIA EVAPORATOR WITH SHORT TUBES

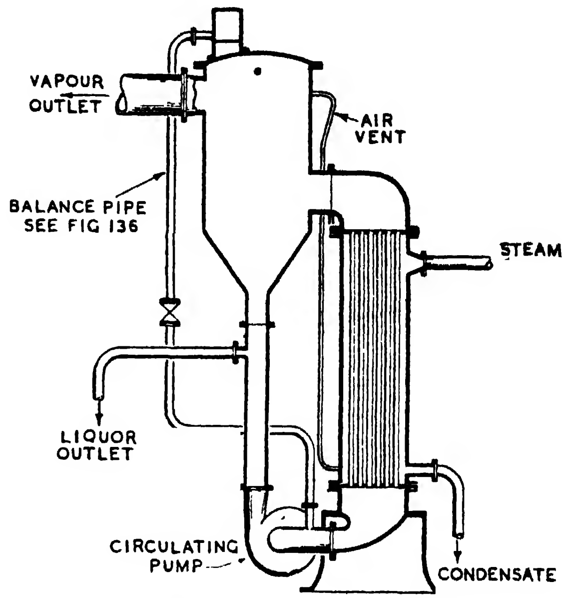


FIG. 188. FORCED CIRCULATION MEDIUM LONG TUBE CALANDRIA EVAPORATOR

solutions in the author's factory. Figs. 187 and 188 show two kinds of mechanical circulator. The design shown in Fig. 188 is positive pumping and can rightly be called forced circulation. The design shown in Fig. 187 can only be called assisted circulation. Evaporators of design Fig. 187 have been made in enormous sizes—up to 36 ft. in diameter.

Mechanical circulation is not without fault in some applications. For example, every time a sugar solution is pumped it increases in colour and a little of the sugar is destroyed. On the other hand those solutions damaged by pumping are also damaged by heat. Quick circulation decreases the time that the liquid is exposed to the heat and this benefit may outweigh the damage done by pumping.

359. FORCED CIRCULATION CRYSTALLISING PANS. As the material in crystallising pans is amongst the least fluid of all evaporated substances it might be thought that here was a great opening for forced circulation. Actually the material is often so tough that true forced circulation is out of the question; the material can only be " persuaded " by an impeller inside the vessel. The author's experience of mechanical circulation in crystallising pans has been disappointing. The power required was enormous, the increase in heat transfer was barely perceptible, and the effect on crystal formation was bad. In fairness,

however, it must be said that the author has seen a number of mechanically circulated pans in operation in U.S.A. and their owners were pleased with their performance. Fig. 189 shows the design most commonly used—Webre.

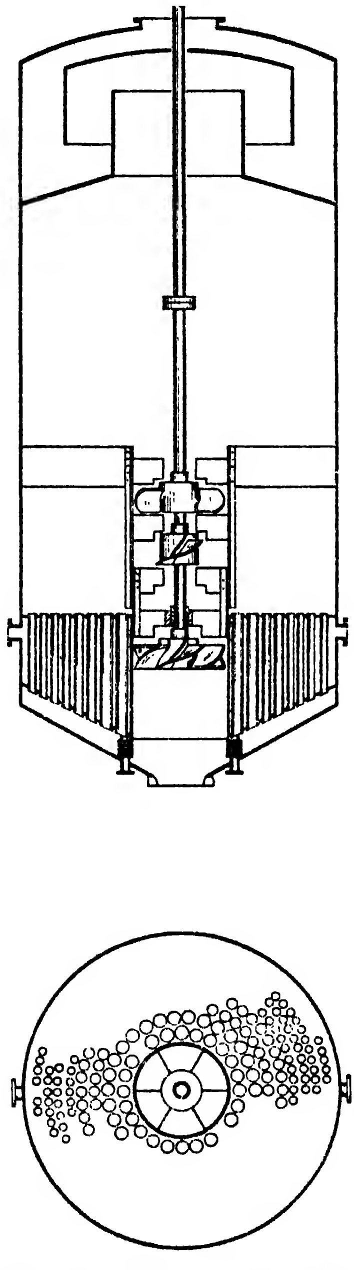


FIG. 189. ASSISTED CIRCULATION CRYSTALLISING PAN

360. NATURAL CIRCULATION PANS AND EVAPORATORS. The great bulk of evaporation is done in natural circulation vessels. Natural circulation is used for many reasons. It is cheap and simple. It is generally entirely adequate provided real care is taken with the design. Mechanical circulation is generally accompanied by some undesirable quality. Natural circulation eliminates a mechanism that can break down and which requires maintenance, and uses a deal of power.

Good circulation is all-important in a pan or evaporator. It is not only necessary for quick heat transfer ; it is generally necessary for good process work. Sluggish circulation permits the material to be overheated, possibly even burnt on to the heating surface. In crystallisation, sluggish circulation allows crystals to adhere giving rise to irregular crystal lumps from which it is difficult to remove the mother liquor, consequently such lumps are never so pure as are single crystals.

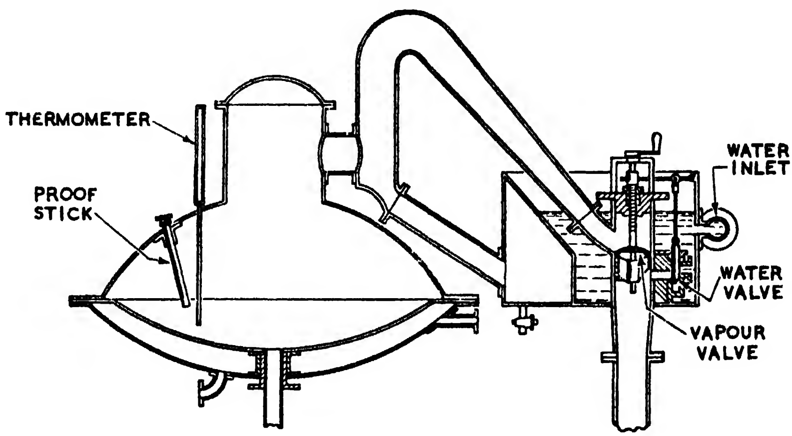


FIG. 190. HOWARD'S VACUUM PAN—1813

The aim of good design in a pan or evaporator should be circulation, circulation and still more circulation. Unfortunately pan and evaporator design is an art, or a trial-and-error business, not a science. It is practically impossible to foretell the performance of a new evaporator of new design or of an old evaporator on a new material, even if the new variations are seemingly small. The only way to find out is to try it. There is little scientific basis for design. The pan and evaporator makers have files of past experience from which they are able to make fair estimates of probable performance provided nothing strikingly new is being introduced.

The following sections give some examples of practical experience gained by the author during the last 45 years. They apply only to water and to sugar solutions, for the most part to very concentrated solutions. As this book is not intended to supply data for the design of evaporating plant this very narrow aspect of the problem probably does not matter. The points brought out are matters of experience and as such are probably of general interest.

361. COIL CRYSTALLISING PANS. The object of a crystallising pan is the production of crystals of a particular size and evenness. Actual rate of evaporation is not the main consideration, though it is important, first because it reduces the size of plant for a given output, and second because the material is subjected to heat for a shorter time if the process is quick.

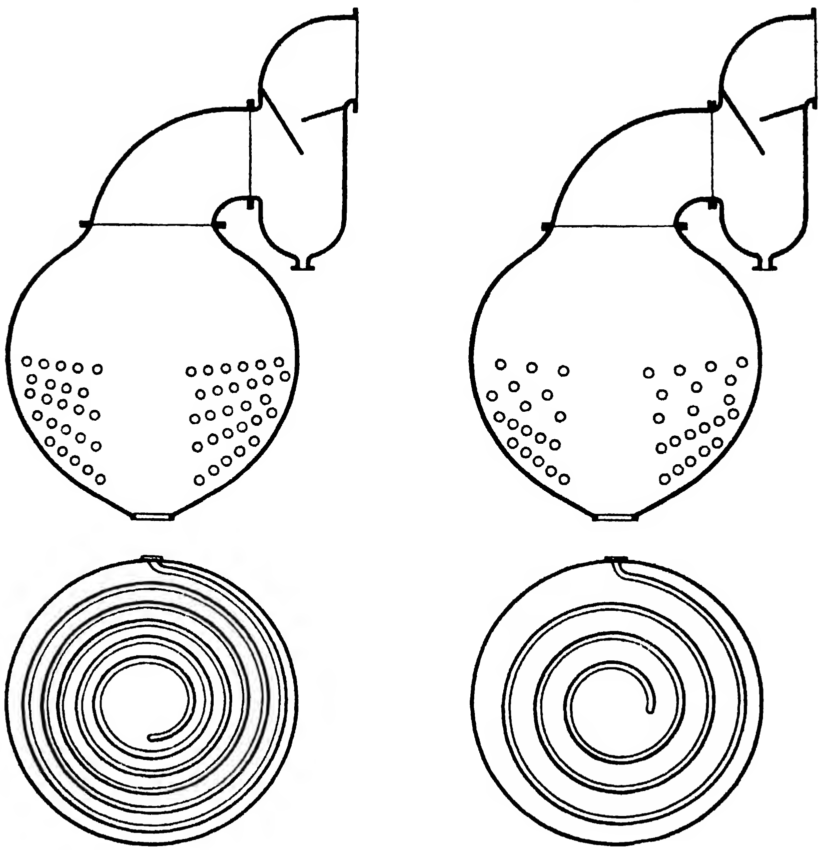


FIG. 191. OLD CRYSTALLISING PAN WITH LARGE HEATING SURFACE AND POOR PERFORMANCE. WHEN THE UPPER HEATING SURFACE WAS REDUCED THE PERFORMANCE WAS IMPROVED

All the older pans were heated by jackets, supplemented by coils as soon as higher performance was wanted. The vacuum pan was invented by Howard in 1813. His vacuum pan is shown in Fig. 190. We get from this picture an interesting insight into his ingenious and economical mind. He used one tank as save-all for his entrained liquor and for his condenser control. He used one control to throttle his main vapour inlet into the condenser and simultaneously to reduce the amount of water admitted to the condenser. A brother

of the twelfth Duke of Norfolk, the Hon. Charles Edward Howard, was a brilliant chemist, engineer and sugar technologist, and made many outstanding inventions which were two generations ahead of their time.

It has already been explained that for very small pans the jacket alone is ideal. As the size increases the jacket becomes less and less adequate and must be supplemented, until in very large sizes the jacket is a relatively small part of the heating surface. The jackets in the larger pans tend to receive less and less care in design as their importance diminishes, and many jackets are just conical sheets riveted to the pan base, instead of the pan being well and truly "embedded" in the jacket. Much trouble with jacket leaks has been experienced in the larger pans.

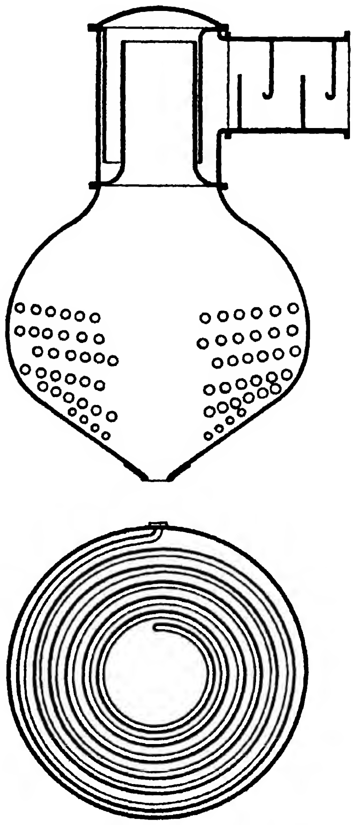


FIG. 192. OLD CRYSTALLISING PAN WITH UNEXPECTEDLY GOOD PERFORMANCE

Coil pans suffer from the disability that the coil must to a greater or lesser extent obstruct the circulation. This trouble of course applies to any form of heating surface other than a jacket, but the coil pan coils are the worst offenders. Much depends upon the shape of the pan and the lay-out of the heating surface, and it is impossible to dogmatise about shape.

Increase of heating surface gives greater total heat transfer—up to a point—but diminishing heat transfer rate. As the surface is increased the obstruction to circulation eventually becomes such as to cause a diminishing total heat transfer. Fig. 191 shows an old crystallising pan installed by the author's father in 1882. Fig. 191a shows the pan as originally installed and Fig. 191b shows some of the coils cut out. The surface volume ratio was reduced from

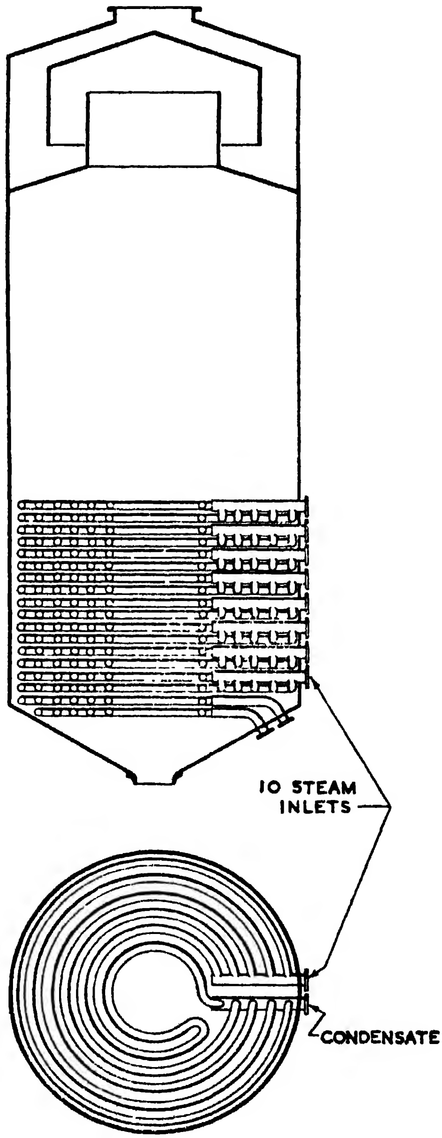


FIG. 193. MODERN CRYSTALLISING PAN WITH BAD PERFORMANCE

1·02 in Fig. 191a to ·77 in Fig. 191b yet the performance with the lesser heating surface was considerably better. To verify this the missing heating surface was restored many years later and a poorer performance resulted.

Fig. 192 shows another old sugar pan—superficially very similar to that shown in Fig. 191. The pan shown in Fig. 192 performed outstandingly well. Why ? No satisfactory reason was ever found. Was it due to the rather elegant turnip shape ?

Fig. 193 shows a pan that was built in 1929. It has a very large heating surface. It had been more or less proved that there was no real virtue in the turnip shape, although the turnip certainly looks as if it would help natural circulation. Probably the reason the " hemispheroids " were used last century was simply tradition coupled with caution against collapse under atmospheric pressure. The pan shown in Fig. 193 was a failure although the surface/volume ratio was large and although the steam pressure used was high. The tangle of " technical worms coiling spirally within " effectively choked circulation.

362. TUBULAR CALANDRIA CRYSTALLISING PANS. Coil pans and calandria pans correspond exactly to fire-tube and water-tube boilers. In a coil pan the heating medium is inside the tubes, in a calandria it is outside the tubes. (The derivation of the word " calandria "—a very pleasant word—is obscure. The French word " calandre " means not only " calender "—both are probably corruptions of cylinder—but also means " shell ".) Calandrias were first developed in the sugar industry for evaporating beet and subsequently cane juices. For many years it was thought that calandrias could not be used in crystallising pans, but when they were tried the results were excellent.

With a calandria all the heating surface is available throughout the life of the boil, whereas in a coil pan only the bottom coils can be used at the beginning ; the other coils being successively brought into use as they become submerged.

Like coils, the calandria offers resistance to circulation, but the resistance is probably less. The material rises in the calandria tubes, which are short, smooth and straight. The material knows exactly what it is supposed to do ; it is not buffeted about and thwarted in its natural movement as it is in a coil vessel. The size of the tubes can be suited to the consistency of the material.

Fig. 194 shows the basket calandria that replaced the coils shown in Fig. 193. This was a great improvement. The pan was, however, by no means perfect. At certain rates of evaporation and height of content it boiled in a series of surges. This was almost certainly due to inadequate circulation allowing the lower material to become overheated. As soon as an ebullition occurred it gave a momentary increase of circulation, the overheated material rose to the surface and there was a great flash of evaporation.

363. RIBBON HEATING SURFACES. In a search after better circulation the calandria shown in Fig. 194 was replaced by the Gräntzdörffer concentric ribbon calandria shown in Fig. 195. Calandria is perhaps a misnomer ; the ribbons are really flat coils. The design shown in Fig. 195 was purely

experimental. The original Gräntzdörffer heating surfaces were made by riveting two plates together and bolting each circular element to the steam connections. This multitude of joints was considered to be out of the question, apart from the difficulty of getting two bolted joints in a 4-in. space. Many years of suffering with bolted and riveted copper heating surfaces had shown that welded steel was greatly preferable. The lower heat conductivity of steel is unimportant compared to the difficulty of getting the heat from the heating surface to the product.

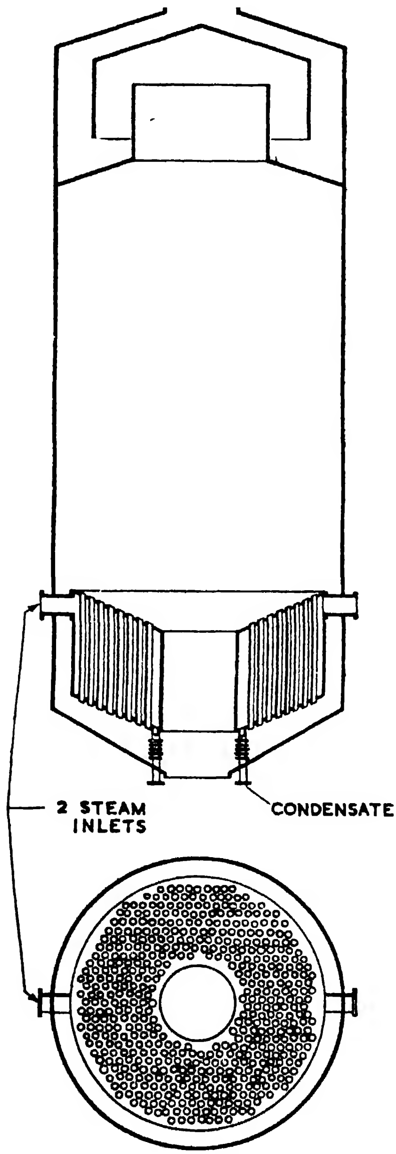


FIG. 194 PAN IN FIG. 193 WITH CALANDRIA IN PLACE OF COILS

This heating surface was a great technical success but from a practical point of view was quite unsuitable. If a stay sprang a leak it was necessary to poke about in a 4-in. space with a 2-ft. electrode—enough to daunt the stoutest welder! But this design showed the way. The heating surface in Fig. 195 is a little larger than the calandria shown in Fig. 194 but the obstruction to

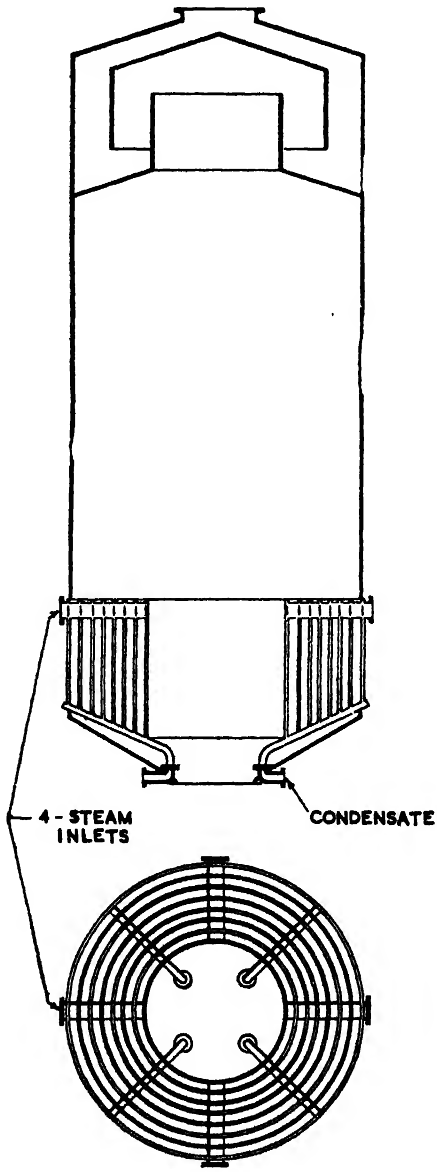


FIG 195. PAN IN FIG. 193 WITH GRÄNTZDÖRFFER CIRCULAR RIBBON HEATING SURFACE

circulation is much less. The obstruction area in Fig. 194 (neglecting the downtake) is 51 per cent.; the obstruction area in Fig. 195 (again neglecting the downtake) is 28 per cent.

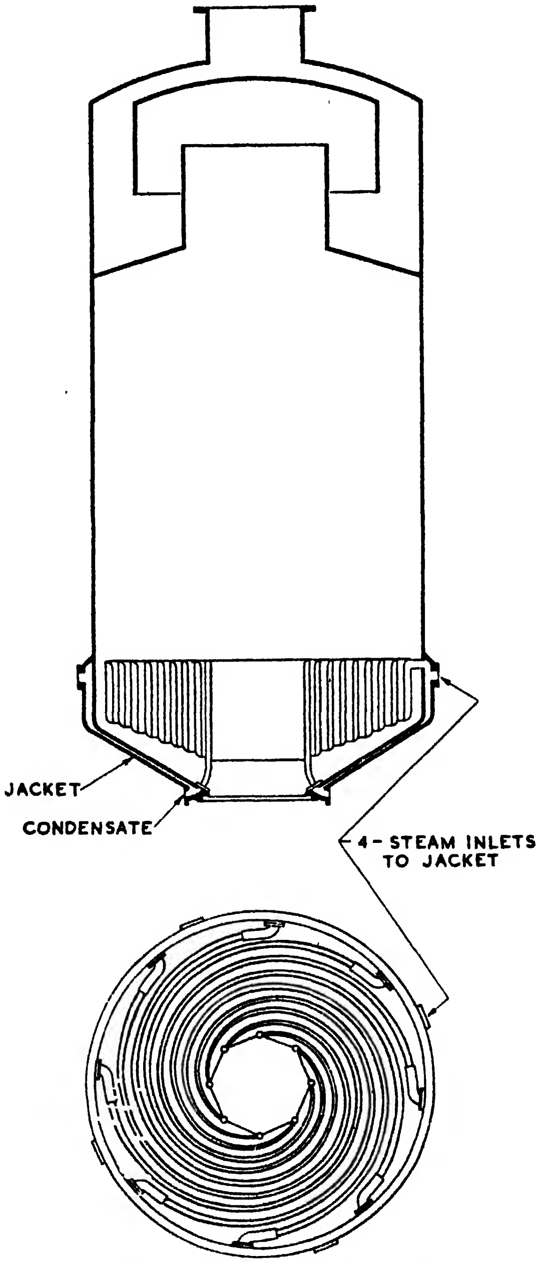


FIG. 196. MODERN HIGH YIELD CRYSTALLISING PAN WITH SCROLL RIBBON HEATING SURFACE

It was therefore decided to design a ribbon heating surface that would be easy of erection and maintenance, and to raise the steam pressure to compensate for any reduction in heating surface that might result. Fig. 196 shows the design which combines a repairable ribbon heating surface with as much jacket as possible. This design only offers 32 per cent. obstruction area. The steam has a definite path to a remote point for venting. There is a good falling path for the condensate. Four such pans have been working with excellent results. The crystal-syrup mass at the end of the boil contains only about 6 per cent. of water and 68 per cent. of the sugar is in the form of crystals. Any attempt to reach such a concentration in a coil pan would pull the coils out, and in a calandria pan such a mass would not run out of the pan—it would just sit on the calandria tube plate. The circulation as observed through the top window is excellent but the heat transfer rate is low—only about 100. This is surprising when the circulation is so patently good. Old bad coil pans have in some cases a heat transfer rate of 150. The performance criterion of a crystallising pan is crystal making, not evaporation, and by this criterion the pans are excellent and by observation the circulation could hardly be bettered. After about 17 years the stays in the elements started leaking and the spiral elements are, in 1957, being replaced by accessibly designed concentric ribbons.

364. EVAPORATORS. Evaporators appear in many different types, tubular, calandria and film. The tubular evaporators are largely used for the distillation of make-up water in the power stations. They suffer from the disadvantages of coil pans. The steam tubes are occasionally " spirally coiling worms " but more often are banks of straight or hairpin tubes. In evaporators the rate of heat transfer is much more important than in crystallising pans. The object of a crystalliser is to crystallise ; the object of an evaporator is to evaporate.

The most common form of industrial evaporator is the " standard calandria " evaporator ; two examples of which are seen in Figs. 133 and 165, Sections 293 and 322. Their use is widespread and much information as to their performance has been gathered, but this is not general information, it is specific to one particular product and one set of conditions.

365. HYDROSTATIC HEAD. When a liquid is heated to the temperature which is called boiling point and which varies with the pressure on the liquid and with the physical properties of the liquid, the vapour pressure at the liquid surface is equal to the overlying pressure. Any further addition of heat will make the liquid molecules fly off as cloud of vapour, see Sections 3 to 14, Chapter 1. Below the surface the liquid has an additional pressure acting on it due to the weight of the liquid above. The boiling point of the liquid therefore rises the greater the depth of liquid. Now a boiling liquid cannot be heated above its boiling temperature, but clearly the boiling temperature is different at every level in the vessel in which boiling is taking place. If circulation is sluggish the lower liquid can be heated up to a much higher temperature than that which can exist at the liquid surface. When this overheated liquid rises to the surface the surplus heat is flashed off as it rises. This may cause surging or bumping as has already been mentioned in Section 362. Bumping or surging is always a sign of indifferent circulation.

There is another force tending to raise the boiling point of the liquid, namely surface tension. The force exerted by the skin of a minute steam bubble is considerable. There is no need to go into the matter in detail ; all we need to remember is that the liquid at the bottom of a vessel can be at a higher temperature than that due to the pressure on the liquid together with the hydrostatic head.

Actually nearly all boiling in a vessel where there is a considerable body of liquid in contact with a heating surface well below the liquid level, takes the form of superheating, rising, flashing. The better the circulation the less the overheating and the smoother the boiling.

Suppose an evaporator contains sugar cane juice at 25-in. vacuum and suppose there is 6 ft. of juice in the evaporator. A 65 per cent. sugar solution has a specific gravity of 1·3 at 140° F. so that there will be a hydrostatic head of 3·38 psi acting on the liquid at the bottom of the evaporator (Table XXXII, Section 206). This additional pressure will reduce the vacuum at the bottom of the evaporator from 25 in. to 18 in. where the boiling point will rise from 140° F. to 175° F. Here is a good argument for maximum circulation and minimum liquid depth.

366. PARTLY SUBMERGED HEATING SURFACE. By keeping the liquid level well below the top of the calandria in a standard evaporator both these requirements are approached. Fig. 197 is intended to show successive occurrences in a single tube of a standard calandria as the liquid commences to boil. At *a* the liquid is standing about one-third of the way up the tube. At *b* boiling has just started. At *c* boiling is more vigorous. The bubble *x* has thin walls which heat quickly and evaporate inwards expanding the bubble. At *d* the bubble *x* has so expanded that it is a thin-skinned sausage of steam. At *e* the bubble *x* has burst throwing drops of liquid across the tube plate. Each tube becomes a steam lift pump just like an air lift. The effect of this action is to maintain a very rapid movement over the heating surface at the same time reducing the hydrostatic head and possible overheating.

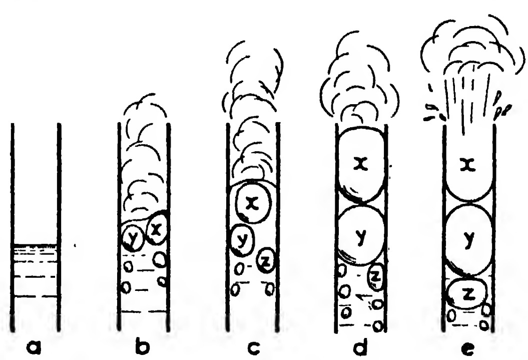


FIG. 197. THE MECHANISM OF BOILING IN CALANDRIA TUBES

Webre has observed that a standard evaporator while heating up was carrying a pressure of 11 psi of steam on the calandria. As soon as boiling started the steam pressure dropped to 5 psi, showing the sudden increase in

heat transfer rate due to the movement induced by the ebullition. Margaret Fishenden observed 20 years ago that ebullition by itself, apart from any induced circulation, increases heat transfer. Almost all standard calandria evaporators work with the liquid level well below the top of the calandria. The best height is somewhere between one-fifth and one-third up the tubes.

The advantage of not submerging the calandria is given by Badger on an evaporator with 30-in. tubes :—

	Heat transfer rate
Calandria covered	400
Liquid 15 in. up tubes	530
Liquid 7 in. up tubes	580

Tests done in the author's factory on water in 1934 on a small standard calandria evaporator did not show such a marked difference. The evaporator had 2 in. tubes 36 in. long, steam pressure 6·5 psi.g. or 231° F. water boiling under 5 in. vacuum or 203° F.

	Heat transfer rate
Calandria covered	540
Liquid 18 in. up tubes	570
Liquid 8 in. up tubes	590

Foust, Baker & Badger measured the natural circulation velocity on water in a tubular calandria at different liquid levels and with varying temperature differences by means of a Pitot tube. They obtained the following results in tubes 48 in. in height and 2½ in. diameter :—

Liquor level above or below top tube plate	WATER VELOCITY AT BOTTOM OF TUBES WITH NATURAL CIRCULATION—FT./SEC.		
	Temperature difference		
Inches	20° F.	30° F.	40° F.
— 12	1·1	1·3	1·4
— 8	1·4	1·8	2·1
— 4	2·0	2·4	2·9
0	2·4	3·1	3·6
+ 4	2·3	3·5	4·0
+ 8	2·0	3·2	4·0
+ 12	1·7	2·7	3·5
+ 16	1·5	2·3	2·8
+ 20	1·3	1·6	2·2

Over the range investigated the H.T.R. steadily decreased as the water level was raised. This does not contradict the principle that H.T.R. increases with velocity, because in the " empty " part of the tube there is a higher film or foam velocity than in that part full of quiet liquid.

367. CLIMBING FILM EVAPORATORS. The extension of the steam lift principle shown in Fig. 197 to a really long tube, 20 or 30 feet, with the liquid standing only a few inches up the tube results in a very high velocity of liquid film being dragged up the tube by the steam, at first as expanding bubbles, further up as a core of rushing steam dragging a tube of liquid up with it (Fig. 198). Heat transfer rates are high and the liquid is only in the evaporator for a few minutes.

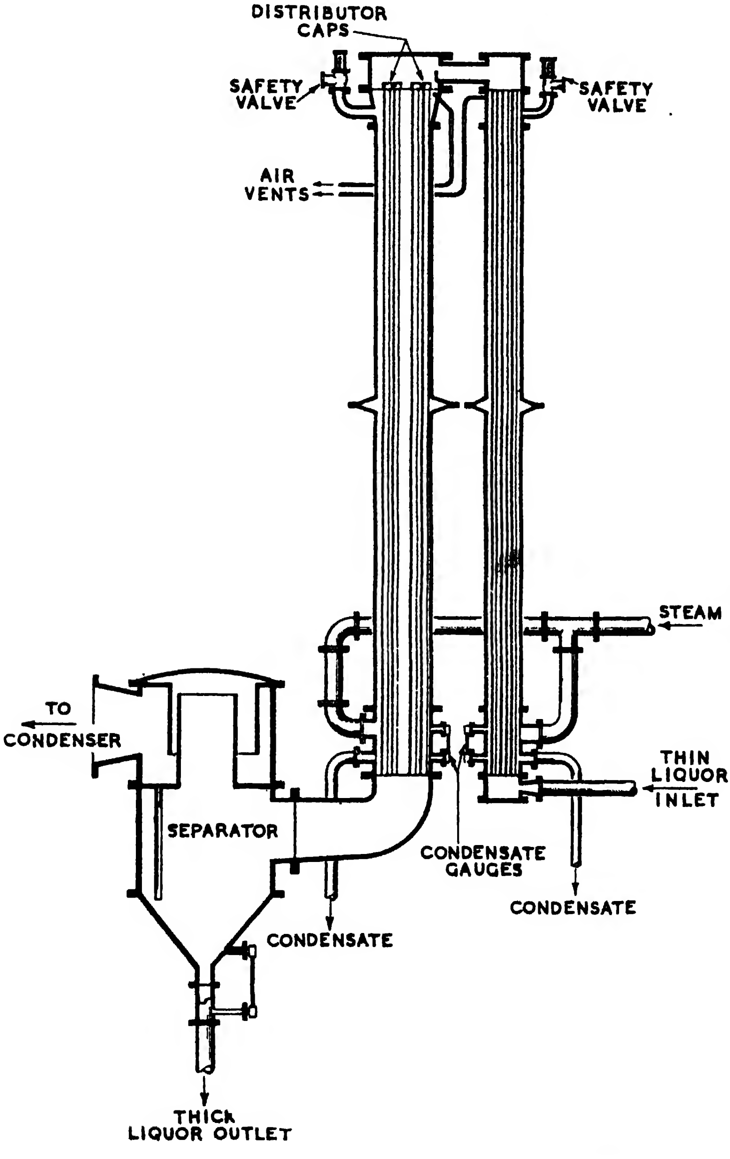


FIG. 198. SINGLE EFFECT CLIMBING AND FALLING FILM KESTNER EVAPORATOR

The climbing film evaporator, however, exhibits some curious unstable traits. If the feed is at boiling temperature corresponding to the vacuum in the vessel the liquid will commence to boil immediately it enters the tubes. Bubbles, or "chaplets" (rosary or string of beads) as Kestner called them, will travel up the tube dragging a film of liquid with them. Evaporation of the film increases the amount of steam until the tube, considered as a steam pipe, is very heavily loaded. This heavy steam flow calls for a large pressure drop over the tube length. The vacuum at the top of the tube will remain constant provided the condenser is man enough to deal with the steam, consequently the pressure at the bottom of the tube will rise. This raises the boiling point. The entering feed is no longer at boiling point so that the liquid level will rise and the lower part of the tube will act as a feed heater. The extra liquid depth puts a hydrostatic head on the liquid raising its boiling point still more so that there is a possibility of overheating. Any attempt to remedy matters by increasing the temperature of the feed will defeat its object, because the hotter feed will evaporate more readily, this will cause an increase in the amount of steam, the tube will be still more overloaded, the pressure drop will increase causing still more rise of pressure at the bottom of the tubes. A climbing film evaporator therefore must not be pushed. It should be operated at such an output as will keep the temperature at the bottom of the tubes below some given point. In this way the amount of liquid in the evaporator will be a minimum and the temperatures will also be minima. The rate of evaporation can be exactly controlled by the steam pressure on the calandria. The unstable condition described above has been deduced from temperature and pressure readings taken at the top and bottom of the vessel ; it has not actually been seen. The glass model climbing film evaporator in the laboratory in the author's factory shows no such condition. The bubbles form, elongate, accelerate, and burst all in accordance with the book of the words. The bore of the glass tube is greater, compared to its length, than that in the full sized plant. There is therefore less pressure drop. This may account for the apparent absence of overloading.

It has been found that this liquid rising condition due to pressure drop can be greatly improved by admitting a little air into the liquid at the base of each tube. The air expands under the reduced pressure and starts things off as an air lift until the steam lift gets going. In the author's factory admission of a little air increased the output of a climbing film evaporator by 10 per cent. and yet was insufficient to reduce the vacuum of 27 in. under which the vessel was working.

368. FALLING FILM EVAPORATORS. No such filling up with liquor can occur with a falling film evaporator, which is a similar long vertical tube evaporator with the liquid feed at the top instead of at the bottom. The distribution of the feed to each tube is carried out in effect, though not in actual detail, by making all tubes proud to exactly the same level above the tube plate. This plant cannot become unstable, there can be no hydrostatic head to raise the boiling point ; gravity, for what it is worth, is acting with the liquid flow instead of against it, and generally speaking the advantages are such that it is surprising that the falling film evaporator has not superseded the climbing film. The real difficulty is, as might be expected, even distribution of liquor feed to each tube,

and even distribution is so difficult to achieve as to handicap the falling film evaporator very seriously.

There is, of course, a pressure drop down the pipe and this is accompanied by a temperature gradient, especially if the plant is highly loaded; in other words, the tubes are often too small. Unless the tubes are fairly heavily loaded there is a risk that some of the tubes may not get their fair share of feed and will over-concentrate the liquid flowing down them. That this does in fact occur, even with heavy loading, is clear from the appearance of the concentrated liquid leaving the evaporator shown in Fig. 198. The output liquid has sometimes a mottled appearance due to varying density refracting the light to varying degree. This varying refraction is transient and disappears after a short time when the liquid reaches an even density by diffusion. Great pains must be taken to ensure an even feed to each tube. This is largely a matter of trial and error and can be helped by providing a good window at the top of the evaporator with good illumination of the goings-on inside.

The smaller the tubes for a given output the easier is it to get even distribution, but small tubes result in a large pressure drop across their length. The ideal plant might well have conical tubes which would maintain a good initial velocity, would prevent overloading with steam at the bottom of the tubes and might make the distribution of the feed easier.

The great advantage of the falling film evaporator is the short time that the liquid is inside the plant. It is more a matter of seconds than of minutes. In the plant shown in Fig. 198 the liquid is estimated to be in the climbing portion for three to four minutes and in the falling portion for from 20 to 30 seconds.

It is possible (experiments in a glass model show promise) that the best arrangement would consist of a small falling film vessel followed by a large climbing film body. The liquid entering the falling film vessel would be large in quantity and low in viscosity making for easy distribution. The steam in the mixture entering the climbing body makes certain that there could be no stagnant liquid column in the bottom of the climbing tubes, thus minimising the chance of instability and overloading of the climbing vessel.

It will be seen that in both the climbing and falling bodies in Fig 198 the steam inlet is at the bottom and the air vent at the top. This arrangement seems to break two rules that have been laid down in Sections 300 and 317; namely, that the steam flow should follow the condensate flow and that air vents should preferably be at the bottom of a steam space with the steam inlet at the top.

The reason for the arrangement in Fig. 198 is, that the most important thing in a climbing film evaporator is to get the film climbing as soon as possible and that there is a temperature gradient on each side of the heating surface. It has been explained that, due to hydrostatic head and to the resistance of the tube to vapour flow, the boiling point of the liquid at the bottom of the climbing body will be higher than at the top of the tubes. There will therefore be a smaller temperature difference at the bottom, where it is most important that there should be the maximum heat flow.

As the steam in the steam space flows towards the remote, air venting point, the partial pressure of the steam will decrease as the air concentration increases, thus causing a falling temperature along the steam space.

Were the steam inlet at the top the coolest steam would be in contact with that part of the tube requiring the highest temperature.

In the falling body the pressure drop inside the tubes is to a large extent counteracted by the increased boiling point of the more concentrated liquid, so that the temperature gradient is less clearly defined. For convenience and design tidiness, the same arrangements of steam inlet and air outlet are used as in the climbing body.

369. PERFORMANCE OF CLIMBING AND FALLING FILM EVAPORATORS. Many conflicting figures are published. The conflict is probably due to their being recorded on different materials at different densities. Laboratory heat transfer rates as high as 1,400 have been published, though no figures approaching this have been given for full sized plants. Here are some figures, for what they are worth, of performances in the author's factory :—

Double Effect Climbing Film Evaporator.

	1st Effect	*2nd Effect*
Material : Dilute impure sugar solution :		
Tube length	23 ft. 6 in.	23 ft. 6 in.
Tube diameter	2 in.	2 in.
Tube number	30	30
Steam pressure	12 psi.g.	6-in. vac.
Vacuum	6 in.	25 in.
Liquor density—inlet—per cent. solids	15 per cent.	
Liquor density—outlet—per cent. solids		70 per cent.
Overall heat transfer rate	200 to 280	

Single Effect Two Body Climbing and Falling Film Evaporator (Fig. 198).

	Preheater	*Climbing body*	*Falling body*
Material : Golden Syrup :			
Tube length	75 ft.	20 ft. 11 in.	20 ft. 9 in.
Tube diameter	3 in.	2 in.	2¾ in.
Tube number	1	9	18
Steam pressure	8 psi.g.	8 psi.g.	8 psi.g.
Vacuum at bottom		11·7 in.	27 in.
Vacuum at top		22 in.	22 in.
Density of feed—per cent. solids	62 per cent.	62 per cent.	68 per cent.
Density of outlet—per cent. solids	62 per cent.	68 per cent.	83·6 per cent.
Heat transfer rate	230	330	190

In view of the viscosity and density (16·4 per cent. water) of the material these heat transfer rates, especially in the falling film plant, are excellent. The vapour velocity, as shown by the huge pressure drop, is altogether excessive. The final vapour velocity at the bottom of the falling tubes is about 600 ft. per second. All the vapour from the climbing body has to pass down the falling body tubes. A separator between the two bodies and a pass-out connection from the separator to the condenser might give a big improvement.

370. HORIZONTAL FILM EVAPORATORS. There are two types: internal film and external film. The Yaryan evaporator is the principal example of the internal film, and the Lillie is the principal example of the external film. In the Yaryan evaporator groups of tubes are connected together in series to form what is in effect a single long tube. Limited amounts of liquid are fed to each tube. The plant then works like a climbing film evaporator with steam bubbles and finally a steam core dragging a film of liquid along the inside of the tubes. There are disadvantages. The first is that possessed by the climbing film plant; namely, the pressure drop along the long pipe. The second is that there is a tendency for a small but relatively thick stream of liquid to flow along the bottom of each tube. The third is the difficulty of ensuring even distribution to a number of horizontal tubes at different levels.

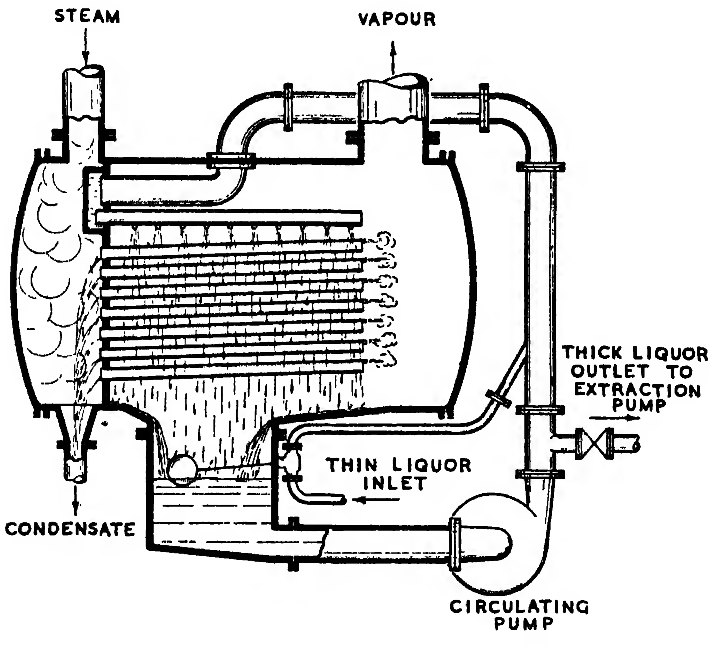


FIG. 199. LILLIE FILM EVAPORATOR

The Lillie evaporator is shown in Fig. 199. The heating surface consists of a bank of tubes supported at one end in one stout tube plate. The tubes are set sloping upwards at an angle of about 5°, and are closed at their upper ends. Steam goes in at the open end and the condensate runs back out of the same end. A small nipple with a hole about $\frac{1}{32}$ in. diameter is fitted to the end of each tube to vent the air. The liquor is circulated by pump and is sprayed over the tubes. The film is thin, there is no pressure drop and there can be no local overheating. The heat transfer rate is high. The author can give no figures from his Lillie plant, but Kerr has given figures as high as 650 at atmospheric pressure. (The heat transfer rate falls off rapidly as the

vacuum increases.) There is more liquid in circulation than in the climbing or falling film machines, and the pump, which may damage delicate products, is a disadvantage to set against the advantages. Probably the principal objection to this type of evaporator, which is shared by all steam tube evaporators, is the difficulty of removing scale from the tubes. Apart from chemical means, annular brushes are the only possible mechanical means and they are most unsatisfactory.

An early Lillie patent in 1888, in which he shows stage vapour feed heating (see Chapter 17) was for a falling film vertical evaporator.

371. COMPARISON OF EVAPORATOR TYPES. Generally speaking, unless there is some good reason to the contrary, the standard calandria evaporator should be chosen. It is cheap, robust, simple, straightforward, easy to control, easy to clean, has pretty good circulation, very little hydrostatic head and does little overheating.

A film evaporator is desirable if the product is so delicate that it must be heated for the shortest possible time. The choice then probably lies between a climbing or falling film evaporator and a Lillie. The falling film has the minimum of material in process but has a considerable temperature gradient and feed distribution is difficult. The Lillie has no temperature gradient but has rather more material in circulation and has a pump.

Horizontal evaporators, which consist of banks of horizontal steam tubes in the liquid, are used in a number of industries, and in particular in power stations for the production of distilled water. With non-viscous liquids very high heat transfer rates are sometimes obtained. The tubes cannot be cleaned very satisfactorily except by chemical means.

Kerr gives the following comparative figures for heat transfer for different evaporators on sugar liquors in quadruple effect. (Multiple effect evaporation is dealt with in Chapter 17.)

TABLE LIII. HEAT TRANSFER RATE IN QUADRUPLE EFFECT EVAPORATORS ON SUGAR CANE JUICE
(Kerr)

	HEAT TRANSFER RATE BTU/SQ. FT./HOUR/°F. DIFF.				
	1ST EFFECT	2ND EFFECT	3RD EFFECT	4TH EFFECT	OVERALL
Standard	205	280	240	40	191
Horizontal	290	237	130	76	183
Lillie	427	427	395	158	351
Kestner	310	325	295	155	277

These figures must only be accepted as indicative of the kind of relationship between the types of evaporators. The inconsistencies in the figures are patent and are probably due to the impossibility of finding evaporators working under comparable conditions.

372. SALT EVAPORATORS. In a very viscous solution like a sugar solution the crystals will only grow very slowly. Due to the viscosity of the liquor the crystals are easily held in suspension and can therefore be grown to any desired size within reason. Some solutions, such as common salt brines, have so low a viscosity that crystals form very quickly and cannot be kept in suspension by the fluid liquid. It is consequently difficult to grow them large, even if large size were wanted. A virtue is therefore made of necessity and the crystallising process is carried out continuously in an evaporator. (Sugar crystals can be kept in suspension in a natural circulation pan until they are over $\frac{3}{8}$ in. long, though occasionally disaster overtakes the operator and the pan " sits down ".)

A typical salt evaporator was shown in Fig. 187. Figs. 200 and 201 show two ways in which the salt crystals can be removed without shutting down the plant. The evaporator has a steeply conical bottom leading to the salt removal gear. In Fig. 200 there are two collecting vessels *A*. These are provided with false bottoms *B* covered with finely periorated screens. A salt discharge door *C* is fitted. The operation is as follows : Valves *E* and *D* are opened to equalise the pressure in vessel *A*. Salt crystals and liquor run into the vessel. As soon as it contains sufficient salt as seen through the sight glass *F*, valves *D* and *E* are closed and valves *G* and *H* are opened. Steam or compressed air is put on the top of the vessel *A* through valve *G* and the liquid is thus driven back into the evaporator, leaving the salt crystals behind on the false bottom. Valves *G* and *H* are closed, the vessel is blown off through valve *I* and the salt is then discharged through door *C*. While one vessel is being discharged the other vessel is filling.

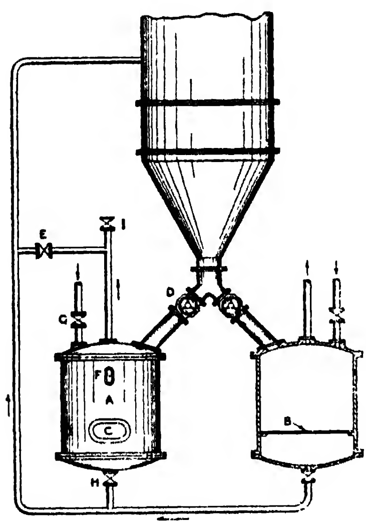


FIG. 200. SALT CRYSTALLISING EVAPORATOR WITH INTERMITTENT SALT DRAW-OFF

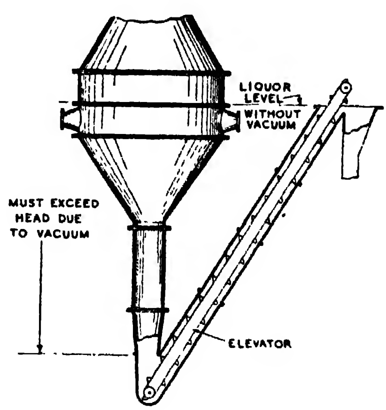


FIG. 201. SALT EVAPORATOR WITH CONTINUOUS SALT REMOVAL

Fig. 201 shows another method which is continuous and automatic. The bottom discharge pipe of the evaporator leads into the boot of an elevator. Elevator and discharge pipe form a large U-tube which seals the vacuum in the evaporator. The elevator and discharge pipe must be long enough to allow of a depression of liquor level in the elevator corresponding to the brine hydrostatic head equivalent to the vacuum under which the plant is working.

373. CRYSTALLISING PANS AND EVAPORATORS. The essential difference between a crystallising pan and an evaporator is that in a crystallising pan the formation of even crystals of a particular size and of maximum yield is the obiect of the process, whereas in an evaporator evaporation is the main consideration.

In many crystallising pans much waste of steam occurs because the pans have been designed with too much emphasis on evaporation. At certain times, particularly when the crystals are very small, crystal growth is very slow and evaporation must be greatly reduced. If there is a large heating surface it may well be impossible to maintain circulation at a very low rate of evaporation. The only course is then to take in water drinks, which are just the same as blowing steam to atmosphere. A crystallising pan should be supplied with liquor that has been well pre-concentrated. The pan should be designed to give good circulation with a small heating surface and a relatively small total heat transfer. The rate of crystallisation in a large pan varies enormously. It has been calculated that, in a 60-ton white sugar pan in the author's factory, crystal growth is at the rate of 1 lb. of sugar in the first minute after crystal growth starts, while it is 1 ton in the last minute before the pan is finished.

374. BREWING COPPERS. A brewing copper is quite a different plant from an ordinary evaporator. Its primary purpose is not evaporation, but simply the extraction of the flavouring and preservative matters from the hops, the coagulation of albuminoid matters and sterilisation. For this purpose a very brisk circulation is wanted and this is obtained by boiling. The evaporation is often quite unnecessary, in fact water is often added to the wort after it leaves the coppers, and the brewing trade are not agreed as to the necessity or otherwise of doing any evaporation at all. Some brewers consider that pressure cooking with mechanical circulation and no evaporation would do all that is wanted.

The steam heated brewing copper, shown in Fig. 234, Section 408, although it has a submerged heating surface, relies for its circulation on the steam lift principle described in Section 366. The heating surface is of the basket type, that is to say wort circulation is up the middle and down the outside. The heating surface in this particular copper consists of a bank of vertical blind tubes like a vertical Lillie. The heating surface is totally submerged but the boiling wort forms a mixture of steam bubbles and wort which rises up the central pipe and flows in an umbrella-shaped stream from the top of the pipe, which is provided with a bonnet to deflect the liquid and prevent entrainment.

Whether there is any real virtue in this type of circulation is an open question. It looks very spectacular, but the actual rate of circulation is low and it is probable that a much faster circulation would be obtained by the adoption of a heating surface more on the lines of a sugar pan. One of the essentials of a brewing copper is very easy cleaning. At the end of the day,

the hops must all be sluiced off the heating surface. This is not easy with the basket type calandria but would be very easy with a ribbon heating surface.

375. WATER DISTILLATION. Water evaporation is the easiest of all evaporation problems. Water cannot be damaged by high temperatures. Water is relatively non-viscous. Very small temperature drops can be used because it does not matter how long water is in process. Any convenient temperature range can be used that fits in with other parts of the plant. The only serious difficulty is scaling of the evaporator heating surface, and it is curious that so many water evaporators are made with tubular heating surfaces which are the most difficult of all to clean, unless the scale can be allowed to get so thick as to be removable by cracking. The tubular evaporator was developed for the ship-board distillation of sea water where tubes can become badly scaled in a matter of hours. Some quick method, however imperfect was needed for descaling. Cracking off was the answer. Unless scaling is very rapid and cleaning must be done very often, more easy and effective cleaning can usually be done to a standard calandria than to tubular heating surfaces.

Very high heat transfer rates are obtained on some water stills. This is chiefly because it is often possible to insert them in a high temperature part of the heat circuit, due to the robustness of water. Some water stills operate with a heat transfer rate of 1,000.

376. REPLACEMENT OF REDUCING VALVES BY STILLS. As has been explained in Sections 74 and 117 a reducing valve does nothing thermodynamically useful. It gives a little superheat to saturated steam—see Section 48 and Table VI. If the steam is wet the wiredrawing may dry it to a small extent, but as was shown in Section 182 it needs a very large pressure drop to do any substantial drying. In the example given in Section 182 a reduction from 100 psi to 30 psi will only evaporate 1·9 per cent. of wetness.

Let us take the example given in Section 182, where steam is reduced from 100 psi to 30 psi and let us replace the reducing valve by an evaporator or still, so that the pressure reduction will give us good distilled water as *quid pro quo* for the otherwise useless entropy increase. The performance of an evaporator is largely dependant on the temperature of the liquid fed into it. There are many ways in which such feed heating can be done. In an industry like the sugar industry most of the heat is rejected in the form of vapour at about 140° F. In other industries, dye works for example, the heat is rejected in the form of hot effluent. In such cases a heat exchanger in the effluent line can heat incoming cold water to within a few degrees of effluent temperature. Most factories can, by a little rearrangement, take more heat out of their flue gases, either by installing an economiser or by getting more work out of their existing economiser (see Section 719) when water to the evaporator can probably be heated for nothing to about 180° F. The feed can also be heated by taking heat out of the condensate that would otherwise cause flash. If the feed is already hot it will boil before it reaches the evaporator, if a condensate heat exchanger is used. To prevent the condensate leaving with excessive heat if the feed is so hot that it cannot absorb all the surplus condensate heat, the condensate can be flashed down to 30 psi and the flash steam added to the evaporator output. Another difficulty with a feed/condensate heat

exchanger is that under varying outputs the performance of the heat exchanger varies. If it is desired to keep the condensate temperature constant without automatic temperature control this can be done at 212° F. by allowing the condensate to flash at atmospheric pressure and collecting the flash heat in a simple spray condenser.

Two conditions will be considered, the first with low temperature feed and the second with warm feed. Fig. 202 shows the arrangement with low temperature feed and a feed/condensate heat exchanger. Fig. 202a shows the conditions when 1 lb. of 100 psi steam is put in. Fig. 202b shows the quantities corrected to give 1 lb. of output 30 psi steam. In both of these arrangements there can only be one particular rate of flow that will give the particular heat exchange shown for the particular heat exchanger. In the diagrams the figures in round brackets ( ) represent total heat and the figures in square brackets [ ] represent latent heat in Btu.

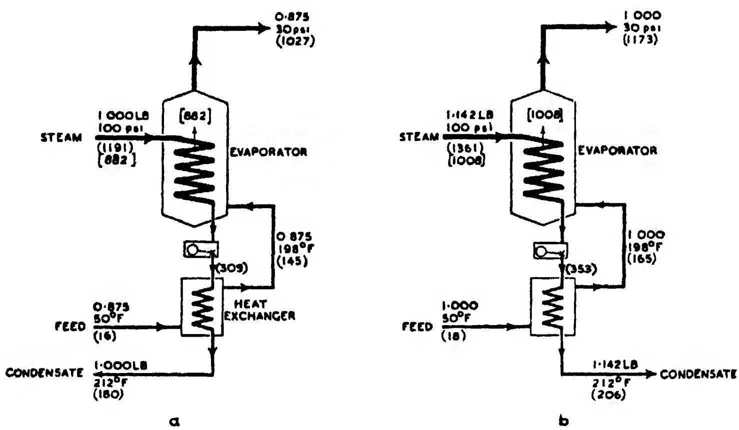


FIG. 202. WATER EVAPORATOR ANALYSIS

The method of working out the conditions in Fig. 202 is :

One pound of 100 psi.g steam put in will give up its latent heat [882] in the evaporator.

The condensate will leave the evaporator with (309) units of sensible heat.

If we wish the condensate to come out at 212° F. it must give up (129) in the heat exchanger.

The total heat given up will be [882] + (129) = (1011).

Saturated steam at 30 psi contains (1173) of total heat.

If the feed is at 50° F. one pound of feed will contain (18) of sensible heat. The heat addition necessary to convert 50° F. feed into 30 psi saturated steam will be 1,173 − 18 = (1,155).

It follows that if 1 lb. of 100 psi steam can contribute (1,011) it will produce

$$\frac{1011}{1155} = \cdot 875 \text{ lb. of 30 psi steam.}$$

The feed therefore is ·875 lb. at 50° F. containing (16) of sensible heat.

In the heat exchanger (129) will be added making the total heat of the feed entering the evaporator (145).

This, on ·875 lb. of water corresponds to (166) on 1 lb. of water, so that the feed temperature will be 198° F.

The latent heat [882] from the 100 psi steam will be added in the evaporator giving (145) + [882] = (1,027) in the output vapour.

Check.—One pound of 30 psi steam contains (1,173) therefore the quantity of output steam containing (1,027) should be $\frac{1027}{1173} = \cdot 875$ lb.

In Fig. 202b the quantities have been raised proportionally to give an output of 1 lb. of 30 psi steam.

In Fig. 203a the feed is taken in at 140° F. The heat exchanger is not applicable, so the condensate is flashed down in two stages, first to 30 psi and then to atmospheric pressure. The 30 psi flash is added to the output and the 212° F. flash is used to heat the feed in a spray condenser.

The method of working out must be a little more complicated.

We put in 1 lb. of 100 psi steam.

This will give up [882] of latent heat in the evaporator, leaving (309) in the condensate.

The condensate now goes to a flash pot at 30 psi. This can either be a vessel with a trap attached or can be a large trap with a top connection to pipe off the flash steam.

One pound of water at 30 psi boiling point contains (243) Btu.

There will therefore be 309 − 243 = [66] excess heat to form flash steam.

The latent heat of 30 psi.g. is [930].

The excess heat of [66] will cause an evaporation of $\frac{66}{930} = \cdot 071$ lb.

The total heat of 30 psi steam is (1,173),

so that ·071 lb. of flash steam will contain 1,173 × ·071 = (83) of total heat.

The flashed condensate at 30 psi will now weigh 1 − ·071 = ·929,

and will contain 309 − 83 = (226) heat units.

This condensate leaves the flash pot and passes into a flash tank at atmospheric pressure.

Now 1 lb. of water at atmospheric pressure cannot carry more than (180) Btu.

so that ·929 lb. cannot contain more than 180 × ·929 = (167).

There will therefore be 226 − 167 = [59] of excess heat to cause flash at atmospheric pressure.

The latent heat of 1 lb. of water at atmospheric pressure is [971],

so that $\frac{59}{971}$ = ·061 lb. of flash steam will be formed.

This will have a total heat of 1,151 × ·061 = (70) Btu.

This flash steam at atmospheric pressure is led into a condenser through which the feed is sprayed.

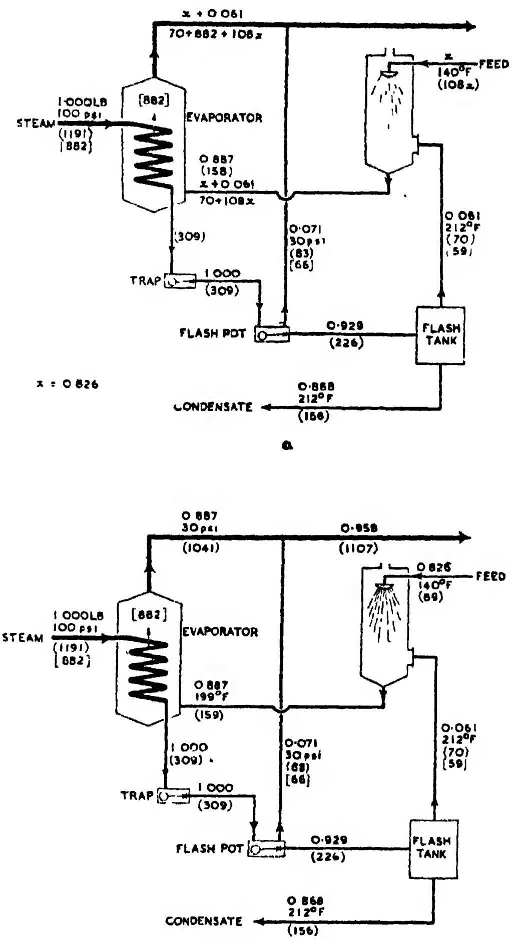


FIG. 203. WATER EVAPORATOR ANALYSIS USING FLASH

We must now find out how much feed can be taken in heated and evaporated.

The feed is at 140° F. and contains 140 − 32 = (108) of sensible heat per lb.

If the weight of the feed is x the feed heat content will be (108x).

In the spray condenser this receives an addition of ·061 lb. containing (70) Btu.

The feed entering the evaporator therefore consists of $x + \cdot 061$ lb. containing $(70 + 108x)$ heat units.

In the evaporator [882] is added so that the output consists of

$$x + \cdot 061 \text{ containing } (952 + 108x) \text{ of heat.}$$

Now 1 lb. of 30 psi steam contains (1,173) Btu so that

$$\frac{1}{1173} = \frac{x + \cdot 061}{952 + 108x}$$

$$952 + 108x = 1173x + 72$$

$$880 = 1065x$$

$$\cdot 826 = x$$

We can now substitute ·826 for x and complete the analysis as in Fig. 203b. We can adjust the quantities all round so as either to put out

1 lb. of 30 psi steam which will call for

1·044 lb. of 100 psi steam,

·862 lb. of feed and give

·906 lb. of condensate ;

or we can produce 1 lb. of condensate which will need

1·15 lb. of 100 psi steam,

·951 lb. of feed and give

1·103 lb. of 30 psi steam.

There are three practical omissions. The first is blow-down, the second is radiation loss, the third is the feed pump. In Fig. 204 all these things have been allowed for.

It has been assumed that a blow-down of 2·5 per cent. is necessary

and that the radiated loss represents about 2·5 per cent. of the total heat input.

The feed pump is assumed to be non-expansive direct-acting, taking steam at 100 psi and exhausting into the 30 psi main.

It is assumed that the overall steam and water efficiency of the pump is 50 per cent. (rather optimistic).

The work the feed pump must do is to pump against the evaporator pressure of 30 psi.g. which Table XXXII tells us is equivalent to 69 ft. head of water.

To pump 1 lb. against 69 ft. is 69 ft. lb. At 50 per cent. efficiency this becomes 138 ft. lb.

Reference to Section 116 tells us that the work available from steam in such a pump is the external energy only — the product of the volume increase and the gauge pressure.

The volume increase of 100 psi steam from water is 3·88 cu. ft.

The external work done by 1 lb. of 100 psi steam is therefore

$$3 \cdot 88 \times 100 \times 144 = 55{,}872 \text{ ft. lb.}$$

In order to do 138 ft. lb. of work the feed pump will use

$$\frac{138}{55872} = \cdot 00247 \text{ lb.}$$

of steam per pound of feed.

Fig. 204a shows the method of working out, and Fig. 204b shows the solution for x.

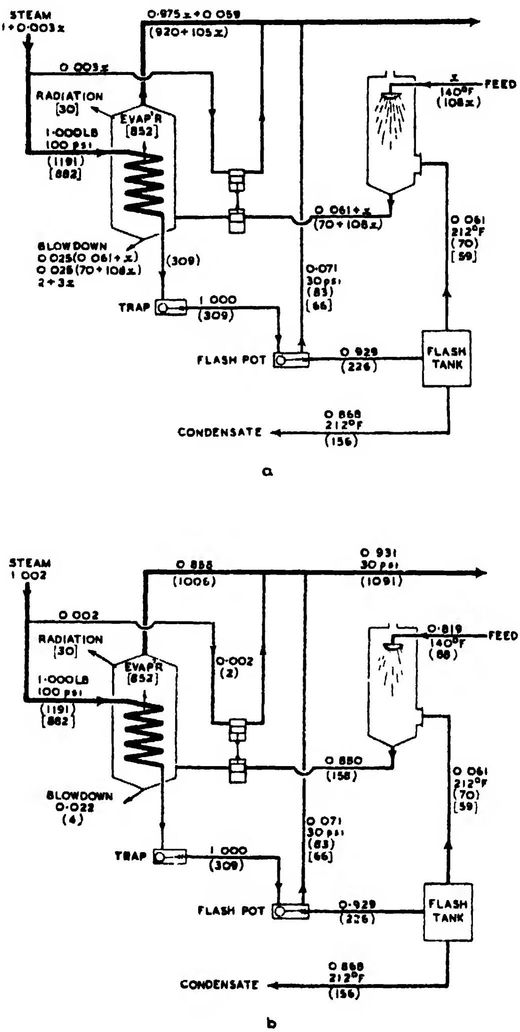


FIG. 204. WATER EVAPORATOR ANALYSIS WITH FLASH USE, FEED PUMP, BLOW-DOWN AND LOSSES

We can now adjust the proportions to make any desired item unity thus :—

	Fig. 204*b*	*A*	*B*	*C*	*D*
Input 100 psi steam ..	1·002	1·0	1·076	1·154	1·223
Output 30 psi steam ..	·931	·929	1·0	1·073	1·137
Condensate	·868	·866	·932	1·0	1·060
Feed	·819	·817	·880	·944	1·0

This is roughly what can be expected in practice. It must be decided whether the distilled water is worth the price that has to be paid for it. Thermally there is no loss except say 2·5 per cent. radiation loss, but there is considerable plant. Let us see what the plant consists of if we wish to produce 5,000 lb. of 30 psi steam per hour.

One trap—10 gallons per minute at 100 psi.
One trap—10 gallons per minute at 30 psi.
One flash tank—say 3 ft. × 3 ft. × 3 ft.
One spray condenser—12 in. pipe 7 ft. long.
One feed pump—10 gallons per minute at 40 psi.
One evaporator—capacity 5,000 lb. per hour.

By scheme B to get 1·0 lb. of 30 psi steam we must put in 1·076 of 100 psi steam.

Fig. 204 shows that the heating surface has to transfer 882 Btu per lb. or input steam.

The total heat transfer will therefore be 882 × 1·076 × 5,000.

The temperature of 100 psi saturated steam is 338° F. and of 30 psi steam is 274° F. so that the temperature difference will be 64° F.

The heat transfer rate will probably be about 500 Btu/sq. ft./hour/° F. diff.

The heating surface will be $\dfrac{882 \times 1{\cdot}076 \times 5000}{500 \times 64} = 148$ sq. ft.

Suppose we say we will use 2 in. tubes for really easy cleaning and make them 36 in. long, then each tube will have a surface of 1·58 sq. ft. If we say that a 12 in. downtake, with a surface of 9 sq. ft. will take care of circulation, then we shall need about 88 tubes. The evaporator will look something like Fig. 205.

This calculation is not given in order to encourage factory owners to design their own evaporators—this is a task for experienced specialists—but merely to enable them to make rough estimates of the kind of plant that will be required for certain tasks.

377. REMOVAL OF SCALE. Scale is deposited on the heating surfaces of most evaporators. Almost all process liquors that require concentration have scale-forming properties. Most scales are mixtures of various salts. One of the most troublesome and most common is calcium sulphate. In some cases in crystallising evaporators the crystals adhere to the heating surface. This occurs with common salt. Vigorous circulation reduces salting-up considerably. Most brines contain calcium sulphate which deposits with the salt. This

makes the removal of the calcium sulphate scale easier though it increases the salting-up because the sulphate cements the salt crystals together. Most salt evaporators require boiling out with fresh water once or twice a week. This sometimes so softens the sulphate scale by dissolving the soluble scale as to allow the sulphate scale to fall off as a sludge.

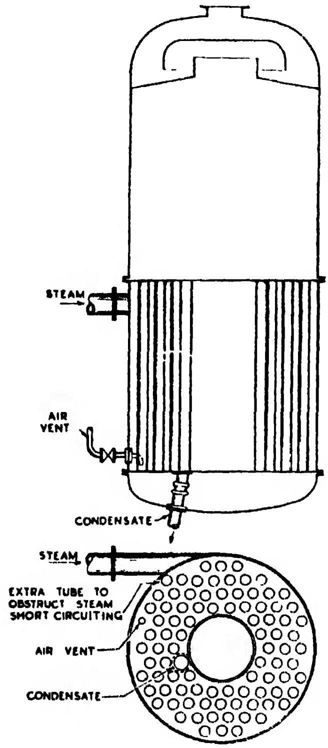


FIG. 205. PRACTICAL DESIGN OF THE EVAPORATOR ANALYSED IN FIGS. 202, 203, 204

There are many ways of removing scale, none of them really perfect. Some evaporators have their tubes slightly oval so that variations in pressure expand or flatten the tubes and flake the scale off. In some factories and industries the tubes are heated with steam with the evaporator empty and the steam is quickly shut off and immediately followed by cold water. The sudden contraction (sometimes) flakes the scale off. In other cases steam is put on to the coils of an empty evaporator and cold water is played on to the coils from a hose through the manhole. This is sometimes successful in flaking off the scale. Sometimes, especially in coil pans, men have to go in and chip the scale by hand. Many power station water stills are made so that the heating surface can easily be withdrawn for greater ease of cleaning.

The standard calandria and the climbing and falling film evaporators can be cleaned with the ordinary tube cleaning tools. They are much more easily cleaned than any form of tubular evaporator, which is, generally speaking, exceedingly difficult to descale, except by chemical means, or, when the scale is heavy and quick growing, by cracking by contraction.

Chemical means are often used. Sodium carbonate or sodium hydroxide separately or mixed are cooked in the evaporator and the hard sulphate scale is converted to carbonate or hydroxide scale which can be more easily scraped or brushed off, or can be dissolved with dilute acid. If the sulphate scale contains an appreciable percentage of scale soluble in hydrochloric acid all that is needed is a soaking in acid. The sulphate is disintegrated if its companion solids are dissolved out. There is little danger to the metal of the evaporator in acid washing provided the acid is dilute. If it is desired to use stronger acid, or to be on the safe side, inhibitors, such as formaldehyde, can be added to prevent the metal being attacked.

One of the best ways of minimising scale formation is to ensure the most rapid possible circulation.

Fluff and dust on an air heater are the same as scale on a liquid heater or evaporator. Dirt is usually an excellent insulator and the performances of many air heaters are disappointing simply because the heating surface has been allowed to get coated with fluffy dust. Air movement does not diminish dirt adhesion in an air heater as liquid movement diminishes scale formation in many liquid heaters. Some forms of dirt are very difficult to remove. High pressure water jets will sometimes be effective, but in many cases cleaning by hand will be necessary. Only too many heaters are built with no provision whatever for access to the heating surface. Every air heating surface will quickly get dirty, must be accessible, must be frequently inspected and when necessary cleaned.

* * *

This Chapter is heavily biased towards sugar industry practice. This was inevitable, but the reader is warned so that he may take generalisations with a grain of—sugar.

* * *

CHAPTER 13

HEATING BY DIRECT STEAM CONTACT

Like a hell broth boil and bubble.
SHAKESPEARE. *Macbeth.* 1601.

STEAM is often used to heat a solid in sterilisers or steamers for cooking or for sterilising, or it may be used to effect a chemical action in autoclaves, as in the curing of rubber.

Steam, blown directly into a liquid, is a very common method of heating. It is simple ; the plant is cheap ; but much steam can be wasted. It is by far the best way of heating water if the steam is clean and the operation is correctly done.

Steam is blown into liquids to effect distillation. The steam lowers the partial pressure (see Section 318) of the liquid and permits distillation to take place at a much lower temperature than would be necessary were the liquid to be left to itself or even to be evaporated under a high vacuum. This steam distillation is very specialised and is only used in industries that have a highly technical staff who know a great deal more about it than does this author. It will, therefore, only be touched on very lightly.

378. AUTOCLAVES, STERILISERS AND STEAMERS. These plants are the same thing, but they go under various aliases. They consist simply of vessels with large steam-tight doors through which the material to be processed is put in and taken out. The material is then treated with open steam at the appropriate pressure to give the desired temperature. The great benefit of this method of heating is that the heating up process is very quick and that an exact, unvarying temperature is obtainable. Saturated, air-free steam can only have one definite temperature at a particular pressure. For sterilisation an exact temperature is absolutely necessary. This temperature must be the germicidal temperature and no more. High sterilisation temperature spoils many things—for example mattresses. Were a heating surface used there would be no guarantee that the temperature shown by the thermometer represented the temperature at all parts of the vessel—in fact it almost certainly would not. Heating up by means of a heating surface would be very slow and the temperature would not remain constant without elaborate thermostatic control. With open steam the temperature instantly rises to that of the steam and stays there ; all parts of the vessel are at the same temperature ; elaborate control is not necessary—but safety precautions are. Autoclaves, sterilisers and steamers have a high accident rate.

Open steam in a kitchen steamer is used for a different reason, although the quick heating and constant temperature are incidental advantages. A steamer is used to cook at moderately low temperatures without losing the juices of the fish, vegetables or puddings. Frying, roasting, baking and grilling are high temperature processes which are not always suitable for vegetables or for invalid food. Stewing and boiling necessarily dissolve some of the soluble matters in the food. Steaming neither dissolves, evaporates, dilutes nor burns the food, and is consequently often a very desirable process.

379. VENTING AUTOCLAVES AND STERILISERS. In all vessels where open steam is heating a solid the whole principle breaks down if the vessels are not properly air-vented. Once the plant is started and the walls are hot there is very little steam flow. There is no definite " remote " point at which to fit a vent. The best way is to fit several vent cocks at different points. Some vents should come from the top of the vessel and some from the bottom. Charge the vessel with material. Open all the vents. Close up the doors and turn on the steam. Allow the vents to blow lustily. Then close all the vents but one and allow each vent in turn to blow full for an appreciable time. There is no use imagining that air does not matter ; it very much does. It may be truly a matter of " air is death ". Suppose a surgical dressings steriliser is half full of air, the pressure is no indication whatever of the temperature, which may be well below germicidal temperature. When post-operative sepsis occurs, no one remembers the air vent on the steriliser.

Suppose an autoclave is used for sterilising a filter cloth that is impregnated with bacteria. It is desired to heat the cloth to 240° F. to kill all the spores. This calls for a steam pressure of 10 psi.g. Suppose the autoclave is closed up and the steam is turned on without any air-venting. When 10 psi is shown on the pressure gauge only 40 per cent. of the volume of the autoclave will contain steam. Consequently the steam will only have a temperature corresponding to the partial pressure of the steam. 10 psi.g. is 25 psi.a. and 40 per cent. of 25 is 10 psi.a. The temperature of 10 psi.a. steam is only 193° F. instead of the desired 240° F. The spores will chuckle in their cosy nest !

In most hospital sterilisers actual germicidal tests are done. A tube of infected cotton wool is placed periodically in the steriliser and the time of processing is adjusted to ensure complete destruction of the bacteria. This is a wise precaution that should never be dispensed with, but proper venting of the steriliser is preferable to trial and error methods.

Sterilising is a time-temperature process, as are certain other contact heating processes. So that effective sterilisation may be secured by extending the processing time at a lower temperature. But an extension of time means a lower output and is therefore undesirable, apart from possible damage to the articles being sterilised due to the extended period under heat.

Another important use for autoclavic methods is in the curing of rubber articles. The vulcanisation temperature is critical. Spoiled work is bound to result if the air is not vented, because the processing temperature will not be reached. The working pressure is probably raised in order to secure correct processing. Then one day the air is vented, by a more conscientious operator, and the work is spoiled by too high a temperature.

Thermostatic air vents can be fitted to sterilisers and to some steamers, but owing to the corrosive action of the steam in some autoclaves—vulcanisers for example—the thermostatic element will corrode away sooner than later. Air venting is however of such importance that it may pay to fit the thermostatic vents and renew them, possibly annually.

380. TRAPPING AUTOCLAVES AND STEAMERS. This presents difficulties. Clearly an expansion trap that will pass air in large quantities is desirable. Any liquid expansion trap is however undesirable on, say, a rubber

curing vessel because the condensate after contact with the rubber will be corrosive and will attack the trap bellows. A liquid expansion trap is therefore less likely to be the choice than a metallic expansion trap. Metallic expansion traps, however, often have a very low discharge rate. It may pay to use a balanced pressure expansion trap and to renew it every year or so.

In the case of food steamers it is only a question of time before the trap gets clogged with fat or fragments of food. It may be better to fit no strainer in front of the trap, but simply a coarse grid to catch the larger food fragments. A fine strainer, unless changed daily or oftener, will choke very quickly. The trap must be very robust to stand very frequent dismantling and cleaning. For this reason a metallic expansion trap is probably best. To enable any form of expansion trap to drain a plant promptly the trap should be connected to the plant by a downward sloping unlagged condensate pipe about 4 feet long.

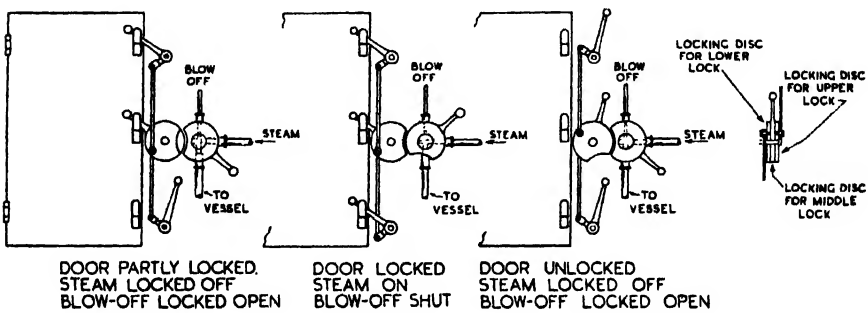


FIG. 206. SAFEGUARDING AUTOCLAVES, STERILISERS OR STEAMERS BY DOOR AND STEAM INTERLOCK

381. SAFEGUARDING AUTOCLAVES AND STEAMERS. The ideal arrangement of door handles and steam valves is such that they are so interlocked that it is impossible to open the door, or unlock one handle, unless the steam is turned off and the pressure inside the vessel is blown off. Safeguards are usually adequately incorporated in the products of reputable manufacturers, but the handles and interlocks may have been removed and replaced wrongly, or a home-made autoclave may be rigged up without proper precautions. Fig. 206 shows a design of locking gear where the steam valve can only be operated when all the door locks are in the locked position. The locks can only be opened when the steam is shut off and the blow-off cock is open. The great disadvantage of any such arrangement is that it calls for cocks instead of valves. Cocks are probably the most troublesome single item of simple plant that exist. Interlocking of valves is much more difficult and much less certain.

382. BLOWERS. Nozzles for blowing steam into a liquid are often called injectors. This is a bad name. An injector is a definite special piece of plant in which kinetic energy is converted into pressure energy and is used for pumping. The author prefers the word "blower" and this will be used.

When steam is blown into a liquid the bubble of steam expands and heats and displaces the adjacent liquid. As the bubble expands some cool liquid comes into contact with it, condensation takes place at once and the bubble may collapse with a bang. Under certain circumstances condensation may occur so quickly that the liquid is drawn back into the steam pipe, from which it is then suddenly ejected. This may set up an oscillation, and a kind of jet propulsion unit is formed which operates with a series of sharp explosions. This will cause disintegration of the plant. Hammering due to badly working open steam blowers will shatter cast iron tank plates, will loosen rivets, fracture welds and transmit undesirable vibration to buildings and to neighbouring plant.

It is essential for the quiet operation of a blower that an even flow of liquid at a steady temperature should mix with the incoming steam. Many designs of blowers are available—some of them serve their purpose excellently. Some designs are shown in Figs. 207 to 210. All work on the ejector principle. The blower shown in Fig. 207 gives good results in small sizes, particularly where it is desired to spread the steam over a long tank. Fig. 208 shows an arrangement that is working satisfactorily in the author's factory, where it is performing the heating whose calculation was done in Section 235, Chapter 6. Endless difficulties over 30 years with this plant had been experienced until many of these small blowers were fitted.

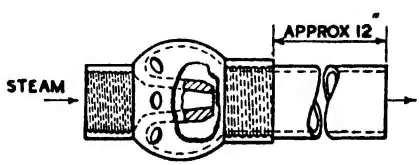


FIG. 207. SIMPLE STEAM BLOWER

Fig. 209 shows a very common design of blower. This design is satisfactory at its rated load (like any injector or ejector there is very little margin) but hammers or back-fires at other loads. If large and varying quantities of steam are to be added, it is essential to split the input up over many small blowers rather than concentrating it in one or two large ones. This spreads the hammerings into a lot of little unsynchronised rattles which may be so small as to be quite unobjectionable.

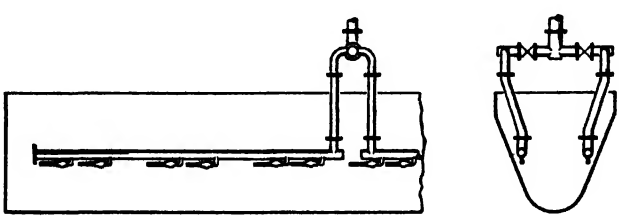


FIG. 208. ARRANGEMENT OF SIMPLE BLOWERS IN LONG CRYSTAL MELTER

Fig. 210 shows a blower that has proved to be fairly satisfactory all-round. It will operate with reasonable smoothness over quite a large load range. It is almost completely silent round about its rated load. It has the advantage that it is designed to be fitted on the end of a drop pipe so that there is no need to bring a steam connection through the bottom or side of the vessel. This is a great advantage where glass-, lead- or other lined vessels are used. It will operate satisfactorily on water or on dirty or very viscous liquids. The small holes A can be added if the material is likely to deposit solids in the saucer top of the blower.

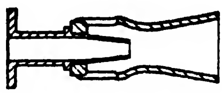


FIG. 209. STEAM BLOWER WITH NARROW RANGE OF PERFORMANCE

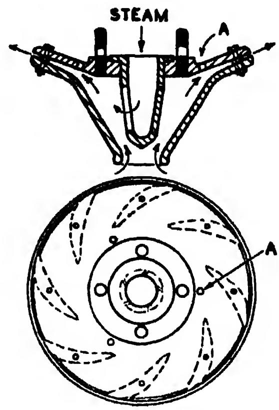


FIG. 210. GOOD DESIGN OF STEAM BLOWER

Blowers can be very unsatisfactory, and, by violent hammering, can cause many breakdowns and much unnecessary maintenance. They will repay care and thought on their choice and installation.

Perforated pipes are often used, sometimes quite successfully. They often fail, however, and sometimes hammer badly. In many cases the material being heated contains solids in suspension or is a material that can crystallise on cooling, or there may be a tendency to throw down a precipitate. In such cases perforated pipes often get blocked and they are very difficult to clean. Where the steam is not condensing in the liquid, as in steam distillation, perforated pipes can be and are being used. Small, widely spaced holes are less likely to cause bumping or hammering than large or closely spaced holes. If a perforated pipe is to be used it should be long with small holes well spaced.

383. THE HAWLEY BOILER. This machine is a simple water boiler which has been in use, at any rate in the sugar industry, for three-quarters of a century. It produces water at just over atmospheric boiling point, and it cannot deliver water either hotter or cooler. It contains no thermostat in the recognised sense of the term.

Fig. 211 shows the original design, of which a number worked in the author's factory for over 50 years. Cold water enters through valve A and is distributed over the surface of the water in the boiler by the perforated pipe B.

The valve A is controlled by the float C which is counterbalanced by the weight D through the chain and wheel E. The steam valve F is controlled by the piston G which is suitably loaded by the weight H. The steam enters through the four tangentially placed blowers K (which hammer like the devil because they are badly designed). When the water is cold there is no pressure in the boiler so that piston G is down and valve F is open. As soon as the water reaches boiling point steam breaks the surface and sets up a pressure in the boiler. This pressure acts on piston G and closes the steam valve F. The water outlet is through the siphon pipe L which is broken by the vent pipe M. No water can get out of the boiler unless there is sufficient pressure to overcome the head P.

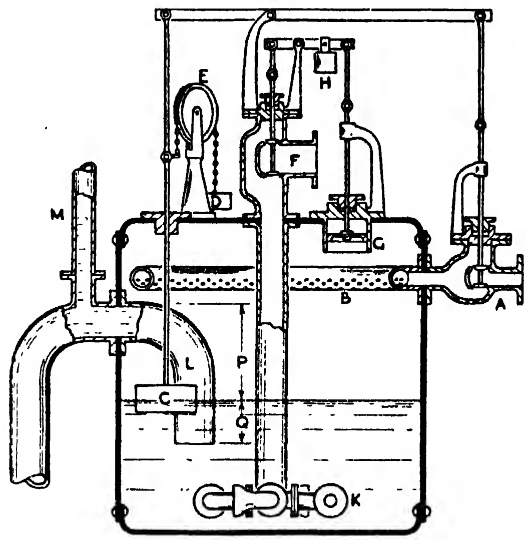


FIG. 211. THE HAWLEY BOILER

When water is drawn off the level drops, float C opens valve A which admits cold water. This cools the boiler contents, lowers the boiler pressure and piston G opens valve F to admit steam to reboil the water and to give the pressure necessary to overcome head P and so discharge water. When the draw ceases the water level rises, float C closes valve A, the steam pressure closes valve F and the boiler shuts itself down until a new draw occurs or until it has cooled enough to drop the pressure, when the piston G admits more steam.

The machine is crude in the extreme and works jerkily. With hard water the piston G eventually jams in its cylinder due to splashes evaporating on the cylinder walls and depositing scale. When the boiler is in a bad state, it occasionally happens that the water level drops quickly with possibly an undue steam admission and the water seal Q may be broken. A mixture of water and steam is then blown out of the vent pipe M. Once this happens it goes on until attended to as it is not a self-correcting condition.

The Hawley boiler could be greatly improved. Gear C E D could be replaced by a float with a bellows seal like the arrangement shown in Fig. 131 Section 292. The water inlet valve could be servo-operated. Piston G should be taken out of the boiler altogether and placed outside with a small pipe connection to deliver the pressure impulse, thus preventing the deposition of scale.

The water delivered by the Hawley boiler must be just above atmospheric boiling point. The pressure inside the boiler before water can be discharged must be atmospheric pressure plus head P. If P is about one foot the necessary discharge pressure will be about ·4 psi.g. when water boiling temperature will be about 214° F.

384. THERMOSTATIC BLOWER TANKS. The Hawley boiler has certain advantages. It cannot, provided it is in good order, waste steam after the water has been brought to the boil, because any steam that breaks surface acts on the piston and shuts off the steam supply. It can never, even when it goes wrong, supply water below boiling temperature. But it may not appeal—it is rather Heath Robinsonian, or, for the sake of the present generation, Emettian. A tank heated by thermostatically controlled blowers has certain advantages. The thermostat can be set to maintain any desired temperature. If however water at boiling point is required thermostat tanks can be very wasteful. The thermostat bulb must be correctly placed and must be very responsive. The hottest water in the tank will rise to the surface, so that the thermostat bulb should be placed in the upper part of the tank well away from the blower.

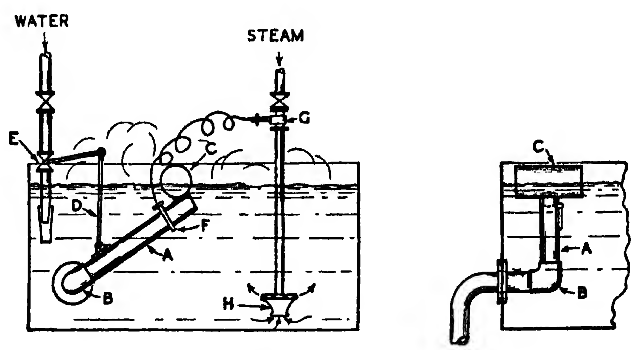


FIG. 212. GOOD ARRANGEMENT OF HOT WATER TANK WITH THERMOSTATIC STEAM BLOWER AND FLOATING DRAW-OFF

Fig. 212 shows the right way of drawing off the water from such a hot water tank, the right way of fitting the thermostat and the right place for the blower. The draw-off pipe A can swing on the loose threaded elbow B so that the hottest water will be drawn off. Float C maintains the mouth of pipe A just below the water surface. The link D operates the water valve E and controls the inlet of cold water which is directed towards the bottom of the tank. The thermostat bulb F is attached to A and so is always measuring the temperature of the water

just below that which is being drawn off. The thermostat operates the steam valve G which admits steam to the blower H.

In some arrangements a modified form of expansion trap is used as a combined thermostat and steam valve. This is shown in Fig. 213 and is clearly undesirable. The thermostat does not control the temperature of the water in the tank, but the temperature round the blower which may well be steam temperature. Another very common fault is also shown in Fig. 213. The tank outlet is placed immediately under the cold water inlet. As cold water is denser than hot water the incoming cold water drops straight into the outlet. The author once lived in a house where the hot water tank was arranged in this way and although the water in one side of the tank was often at boiling point it was never possible to have anything but a tepid bath.

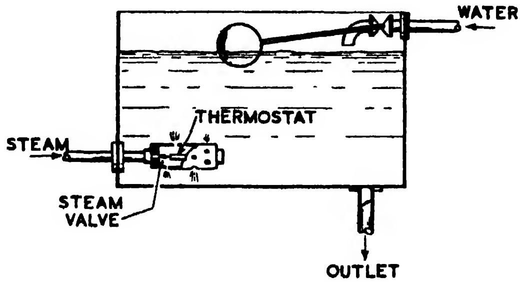


FIG. 213. BAD DESIGN OF HOT WATER TANK WITH THERMOSTATIC BLOWER AND DRAW-OFF BELOW COLD WATER INLET

385. **WIREDRAWING AND WASTE.** When steam is blown into a liquid through a blower the steam must expand. Often the expansion is considerable. Steam at 20 psi may be blown into a tank or vat containing only 3 or 4 feet of liquid. This corresponds to a pressure of only 1 or 2 psi. Saturated steam at 20 psi contains 1,168 Btu. At 2 psi saturated steam has a temperature of 219° F. Steam at 2 psi containing 1,168 Btu has a temperature of 249° F. There will therefore be a superheat of 30° F. at the blower's mouth. Before this steam can condense it must give up its superheat. If the bubbles are fairly large it may not be able to desuperheat itself before it breaks the surface. This often causes great waste—particularly in dye vats. See Sections 48 and 176.

It has been explained in Section 182 that the wiredrawing of wet steam will effect some drying of it, so that it might be thought that as process steam is almost always wet, there would be no superheating from wiredrawing. This is not necessarily true when the wiredrawing is taking place submerged in a liquid. If the moisture is present as actual water droplets these may be scrubbed out of the steam in the blower, leaving the steam dry as it emerges from the blower. It will then expand and superheat itself.

The drying effect of scrubbing wet steam with water is used in the steam drums of modern high pressure boilers for reducing the moisture in the steam delivered to the superheater. The steam is made to bubble through some of the water by means of a series of baffles and the drying effect is very marked.

(87664) M*2

It is therefore important that steam that is to be blown directly into water or a water solution should be reduced to the lowest possible pressure as far upstream of the blower tank as possible, so that any superheat can be used up by making up radiation losses and the steam will reach the blower at the lowest possible pressure and with the least amount of superheat. This will probably be quite successful. It has been explained in Section 182 that the wiredrawing of wet steam is seldom sufficient to dry out more than a small trace of moisture. Early pressure reduction is the sure way of minimizing wiredrawing in a blower.

386. STERILISING BY OPEN STEAM. Many organic liquids contain bacteria which may be destructive and harmful. If the liquid is cold water, sterilisation by means of ozone is probably the cheapest method. Ozone may be inadmissible in other liquids, which are therefore sterilised by means of heat. Bactericidal temperature may damage the product, and the choice is between damage by bacteria and damage by heat. It is sometimes possible to kill most of the bacteria by the use of open steam in such small quantity as not to overheat the liquid. Open steam must always be at or above the saturation steam temperature and momentarily steam at this temperature contacts the bacteria. Provided a good circulation is maintained most of the bacteria get a chance of being slain.

In the author's factory warm, dilute sugar solutions, ideal homes for breeding bacteria, are to a great extent sterilised by blowing in open steam which is a much quicker process than heating through a surface. Two samples of infected liquor were heated up by the same amount in the same time ; one sample was heated through a heating surface and the other by means of open steam. Here are the bacterial counts which show the marked advantage of open steam.

	Time	*Temp.*	*Mesophils/m.l.*	
	Min.	*°F.*	*Open steam*	*Closed steam*
Original liquor	0	130	6,400	6,400
	5	185	10	320
	12	185	2	8

It will be noticed that closed steam does the sterilisation after a time, and of course it has the advantage that it does not dilute the liquor.

387. STEAM DISTILLATION. Boiling, as has been explained in Sections 11 to 13 in Chapter 1, depends upon having sufficient energy in the liquid to cause the vapour pressure, exerted by the molecules trying to escape, to overcome the pressure acting on the liquid. If the vapour pressure of the liquid is high it can be reduced to a low partial pressure by blowing steam through. The steam exerts a large partial pressure and greatly reduces the temperature needed to effect vaporisation of the higher boiling point liquid.

The steam, if it is saturated, should not be called upon to provide any of the heat needed to effect vaporisation of the other liquid, because this would introduce water in the liquid phase which would affect the distillation adversely The heat in the distillation steam is therefore all wasted in the condenser,

though it may be possible to recover some of the heat by some regenerative heating process. Thermally, therefore, the process is a bad one, but it may be very practical and good from a production standpoint. Again we can take the opportunity of pointing out that in a process factory the process comes first. Many materials are damaged by high temperature and if the boiling point of the substance that is to be distilled is very high, the only alternative to steam distillation is distillation at very high vacuum, which may be even more costly. Steam distillation is very simple and straightforward. Distillation under very high vacuum calls for tireless maintenance and effort. The costs and disadvantages must be balanced and a decision made for each individual case on its own merits. Probably steam distillation will be combined with vacuum distillation.

388. QUANTITY OF STEAM USED IN STEAM DISTILLATION. A simple example will be considered to bring out some of the points. Suppose it is desired to distil aniline. Under atmospheric pressure aniline boils at 356° F. If we say that there should be a temperature drop of 60° F. across the still heating surface, the heating steam would need to have a temperature of 416° F. and would therefore have to have a pressure of 280 psi.g. This is high pressure live steam. Suppose there is a shortage of live steam but a surplus of exhaust steam, we can use exhaust steam for steam distillation even if it means using more steam. Here is the oversimplified calculation. (The oversimplification is due to neglect of certain complications such as the mutual solubility of aniline and water, etc.)

Annexe B shows vapour pressures to which we shall have occasion to refer :—

ANNEXE B

Temperature		*Vapour pressure—mm Hg.*	
°C.	*°F.*	*Water*	*Aniline*
50	122	93	2·4
60	140	149	5·7
70	158	234	10·6
80	176	355	18·0
90	194	526	29
100	212	760	46
110	230	—	69
120	248	—	97
130	266	—	146
140	284	—	204
180	356	—	760

Suppose it is desired to distil aniline at 140° F.

At 140° F. the vapour pressure of aniline is .. .. 5·7
At 140° F. the vapour pressure of water is 149

The sum of the two partial pressures is .. .. .. 154·7 mm Hg.

This is the pressure of steam distillation
and is equivalent to .. .. 3 psi.a.
or a vacuum of .. .. 24 in. approx.

The amount of steam used for steam distillation is given by the following relation :—

$$\frac{\text{Weight of Steam}}{\text{Weight of Aniline}} = \frac{\text{(Vapour Pressure of Steam at 140° F.)}}{\text{(Vapour Pressure of Aniline at 140° F.)}} \times \frac{\text{(Molecular weight of Steam)}}{\text{(Molecular weight of Aniline)}}$$

$$= \frac{149 \times 18}{5 \cdot 7 \times 93}$$

$= 5 \cdot 1$ lb. of steam per lb. of aniline distilled.

To this must be added the steam needed to provide the latent heat of vaporisation of the aniline. Some of this heat can be taken from any wiredrawing of the steam, or, if the steam were superheated, the superheat can be used for the aniline latent heat. Too much heat, however, must not be taken out of the steam lest condensation occur and introduce another complication.

Now it is clear from the above relation between the steam needed and the vapour pressures, and from Annexe B, that the higher the temperature the less will be the ratio of the vapour pressures, consequently less steam will be used. When the temperature reaches the boiling point of aniline no steam will be needed. Let us see the effect of raising the temperature of the steam distillation to 212° F.

At 212° F. the vapour pressure of aniline is	46
At 212° F. the vapour pressure of water is	760
Pressure of steam distillation	806 mm Hg.
or	15·6 psi.a.

The amount of steam will be $\frac{760 \times 18}{46 \times 93} = 3 \cdot 2$ lb. of steam per lb. of aniline.

Clearly by using steam distillation we can distil aniline at low temperature. The steam needed is considerable, especially if the temperature is low, but the vacuum needed is quite modest and readily got in the factory. If we use 5·1 lb. of steam per lb. of aniline, we can distil at 140° F. under a vacuum of 24 in. At what temperature would aniline distil without steam at the same vacuum of 24 in.? The annexe shows that aniline has a vapour pressure of 155 mm Hg. at about 267° F. To get aniline to distil at 140° F. without steam would mean evaporating under a vacuum of 5·7 mm absolute which will be very expensive and exceedingly difficult to carry out on a large scale.

The most fruitful use of steam distillation is for the removal of small quantities, sometimes minute traces, of not very volatile substances from products that will not tolerate high temperatures, e.g. the deodorising of fats.

It was seen in the aniline example given above that the higher the temperature at which steam distillation is done the less steam is used, but in many cases this is not the whole story. If the object of the distillation is the removal of small quantities of high boiling point substances from a large quantity of product, the use of a higher temperature of distillation in order to save distillation steam will call for more heat to heat the large amount of product up to the higher temperature. If therefore the amount of the volatile substance to be distilled is small it may well pay to use a relatively large amount of distillation steam and save the steam that would have been needed to heat large quantities of the main product up to a high temperature. It is impossible to be categorical. Each case must be worked out on its own merits.

389. INJECTORS. Injectors are used in locomotives, cranes, portable boilers and sometimes on stationary boiler plants for feeding water into the boiler.

Fig. 214 shows the elements of an injector of the simplest possible form.

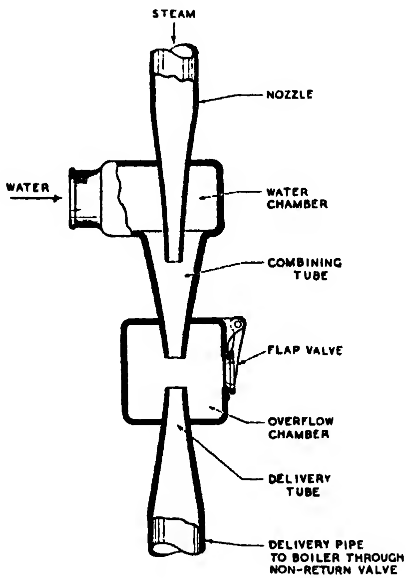


FIG. 214. THE PRINCIPLE OF THE STEAM INJECTOR

If steam is blown through the nozzle, it blows with great speed through the combining tube and escapes through the flap valve of the overflow chamber. Its high velocity creates a suction in the throat of the combining tube and it draws air out of the water chamber until it draws water into the combining tube. As soon as water reaches the combining tube the high velocity steam condenses in the water. The combining tube is narrowed down so that as the steam condenses and the mixture occupies less space, its velocity is maintained.

The high velocity water jet shoots across the gap in the overflow chamber and enters the delivery tube which gradually diverges. The water slows down in the delivery tube and its velocity head is converted into pressure head so that it emerges at low speed and high pressure. As soon as the water jet jumps the gap in the overflow chamber it causes a suction there and closes the flap valve. The various nozzles and tubes are so designed that the water emerges at a higher pressure than the boiler pressure so that the water can enter the boiler.

It is clear that the injector can only operate with cold water as the water must condense all the steam used. As a pump the injector is very inefficient. Its efficiency as a steam pump is only about 1 per cent. On the other hand, apart from a very little radiation and a little loss of steam at the start, the whole of the heat in the steam used is returned to the boiler and it has a thermal efficiency of almost 100 per cent.

The injector is often looked on as an exceedingly inefficient machine and is sometimes thrown out in favour of a feed pump. If for any reason it is impossible to use hot feed water, or if the feed water is heated by feed pump exhaust only, then the injector is generally a better tool than the feed pump. But if the feed water is or can be heated then the injector should give way to a feed pump.

In practice injectors are more complicated than the simple arrangement shown in Fig. 214. The position of the steam nozzle is adjustable with reference to the combining tube.

It is possible, by suitably designing the shapes of nozzle and tubes to use exhaust steam for a boiler feed injector. Much more steam is used, consequently it is even more important that the water be cold. Such an injector is a great improvement on any feed pump using live steam.

Injectors are not to be despised. They are the most efficient feed water heaters it is possible to find. If live steam, or even exhaust steam, is to be used for feed heating the injector may be quite the best way of feeding the boiler and heating the feed simultaneously.

Injectors have another quality which is sometimes an advantage and sometimes a disadvantage. They have a very narrow working range. Generally, they must be either full on or shut off. If the boiler to which an injector is fitted is on a steady load, the intermittent draw of steam by the injector and the big inflow of cool water are disadvantages. But if the boiler is meeting peaks with periods of no steam demand, as in cranes, pile-drivers or shunting locomotives, the big, occasional steam draw by the injector can be used beneficially at times of no load to prevent wasteful safety-valve blows.

390. EJECTORS. Ejectors are used as simple vacuum pumps. For producing very high vacuum they are much more economical than mechanical vacuum pumps, and in fact can produce much higher vacuum than is possible with a reciprocating vacuum pump. They operate most efficiently when compressing air over a fairly small range, about three-fold compression, with a maximum of about six-fold.

An ejector consists simply of a rather simpler injector and the arrangement is shown in diagram form in Fig. 215.

A steam jet blows into a throat and the high velocity steam jet produces a suction which draws the air into the throat at high velocity. The throat then expands and the velocity head is converted into pressure head and the steam and air emerge at low velocity and higher pressure. For economical working the steam pressure should be high.

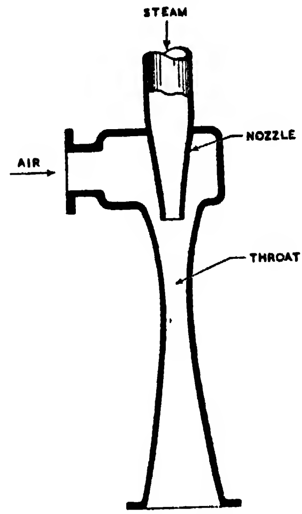


FIG. 215. THE PRINCIPLE OF THE STEAM EJECTOR

In a single stage the ejector will not give much more than about 20 in. vacuum economically if it is exhausting to atmosphere.

In order to achieve a high vacuum economically it is necessary to work several ejectors in series as shown in Fig. 216.

In Fig. 216 the top right-hand ejector draws air from a plant that it is desired should be at 29 in. vacuum. The ejector exhausts into a feed water heater at 27 in. vacuum where the steam condenses. The air at 27 in. from this feed water heater goes into another ejector which compresses the air to 20 in. vacuum into another feed water heater. The air from this passes into a third ejector which compresses the air to atmospheric pressure into a third feed water heater.

Each ejector compresses the air to about one-third of its previous volume.

The principal disadvantage of ejectors is that a use should be found for the exhaust steam. In a power station this is easy, but in a process factory there is usually ample low-grade heat and there is often no use for ejector exhaust whose heat has to be wasted.

The advantages of the ejector are that it is very cheap and small. It can easily produce a much higher vacuum than that obtainable from a mechanical pump. There are no moving parts to go wrong.

For producing high vacuum in a process factory it is often best to use an ejector for the first stage from say 29 in. or 28·5 in. to 27 in. or 26 in. and then come down to atmospheric pressure with a mechanical pump.

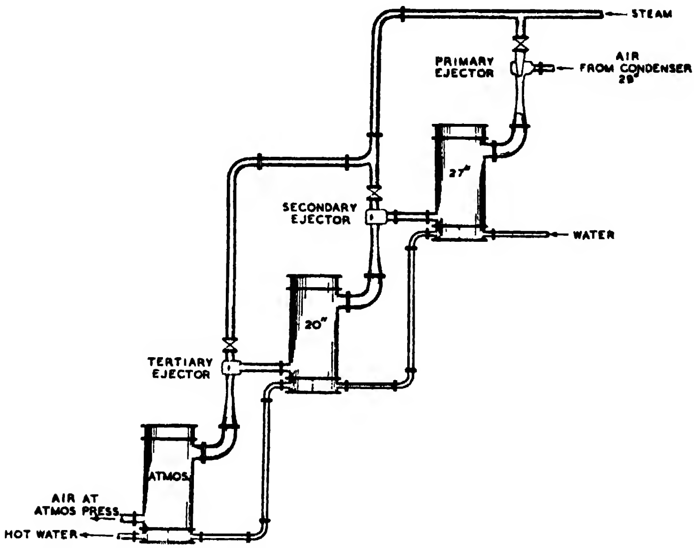


FIG. 216. HIGH VACUUM PRODUCTION BY THREE-STAGE EJECTION

The ejector is used on most British locomotives to produce the modest vacuum required for the vacuum brake. A double ejector is used, not two in series but two in parallel. One large ejector to pull the vacuum up quickly after a brake application, and a smaller ejector to maintain vacuum. An ejector is shown in Fig. 159, Section 320, for producing vacuum in a de-aerator.

* * *

CHAPTER 14

FLASH STEAM AND LOW PRESSURE VAPOUR

They pour down rain according to the vapour thereof.
JOB—XXXVI 27—B.C. 450

The production of flash steam and its useful recovery have already been mentioned in Sections 44 and 46. The question of flash steam is very important and will be dealt with at some length. Before reading this Chapter the reader is asked to refresh his memory by rereading Section 44 in Chapter 1.

391. REASONS FOR MINIMISING FLASH. In many situations it will hardly pay to collect flash steam. Suppose a trap is draining a steam pipe at 50 psi. Apart from the heavy condensate load at the start-up, the discharge from the trap may be only 10 or 20 lb. per hour. If this condensate is allowed to flash to atmosphere Fig. 3 and Table IV show that 9 per cent. of the condensate will be flashed into steam. If 1,000 lb. of steam costs about 10*s.* the value of this flash will be between £1 and £2 a year on one shift, or between £3 and £7 a year on three shifts. Unless this flash steam can be collected very cheaply with a minimum of piping, it clearly will not pay to collect it.

This argument will be used for place after place all round the works until in aggregate there is a very big loss occurring.

If we were able to use 25 psi steam instead of 50 psi, the flash loss would be reduced to five eighths of the 50 psi loss. If we could use steam at atmospheric pressure, or at 1 or 2 psi there would be no flash loss.

So here we have good reason for reducing the number of trapping points to a minimum, so that there are not a multitude of little traps discharging a mere trickle of condensate ; for bringing the traps together into groups as far as is possible ; for lowering the steam pressure to the greatest possible extent.

392. HIGH VERSUS LOW PRESSURE STEAM. If we ignore heat transmission rates, which vary with the temperature at which heat transfer takes place, the whole question of high versus low steam pressure for process or other heating centres round the question of flash steam. If the flash steam, and possibly the condensate also, has to be wasted because the place where the trap discharges is too far away to warrant the expense of piping back, then clearly the lowest possible steam pressure should be used. If, however, there is a use for the flash steam close to the point where it is formed, it would not appear to matter from a heat saving point of view what pressure is used. There is much to be said for and against high pressure heating steam. Let us look for a moment at the way in which the heat in steam at various pressures splits itself up in use. Table LIV shows the heat split for 1 lb. of steam at various pressures.

Now suppose we say that we have a heating job that requires 1,000 Btu. Only the latent heat is given up in the heating plant so we can rewrite Table LIV to show what the heat distribution will be when there is a constant latent heat. This is shown in Table LV.

This shows that at 200 psi we have to put in 11 per cent. more steam than at 25 psi, and that $3\frac{1}{2}$ times as much heat is blown away in flash. At high pressures we can use a small steam pipe to take the steam to the plant, but we need a large pipe to bring the flash back and we recover less condensate.

TABLE LIV. HIGH v. LOW PRESSURE HEATING STEAM—CONSTANT STEAM QUANTITY

STEAM QUANTITY LB.	STEAM PRESSURE PSI.G	TEMPERA-TURE° F.	TOTAL HEAT	LATENT HEAT	HEAT IN CON-DENSATE	FLASH HEAT AT 212° F.	FLASH PER CENT. INPUT	HEAT IN CON-DENSATE AT 212° F.
1	200	388	1,200	838	362	216	18·8	146
1	100	338	1,191	882	309	153	13·3	156
1	50	298	1,180	912	267	103	9·0	164
1	25	267	1,170	935	236	66	5·8	170
1	10	239	1,161	953	208	33	2·9	175
1	5	227	1,156	961	195	18	1·5	177
1	Atmos.	212	1,151	971	180	0	0	180

TABLE LV. HIGH v. LOW PRESSURE HEATING STEAM—CONSTANT HEATING

STEAM QUANTITY LB.	STEAM PRESSURE PSI.G	TEMPERA-TURE °F.	TOTAL HEAT	LATENT HEAT	HEAT IN CON-DENSATE	FLASH HEAT AT 212° F.	FLASH PER CENT. INPUT	HEAT IN CON-DENSATE AT 212 °F.
1·192	200	388	1,432	1,000	432	258	18·8	174
1·135	100	338	1,351	1,000	351	174	13·3	177
1·098	50	298	1,292	1,000	292	113	9·0	179
1·069	25	267	1,252	1,000	252	71	5·8	181
1·050	10	239	1,219	1,000	219	35	2·9	184
1·040	5	227	1,203	1,000	203	19	1·5	184
1·030	Atmos.	212	1,185	1,000	185	0	0	185

The object of taking steam to a plant is to take heat to the plant. The higher the pressure the less heat does a given weight of steam give up to the plant but more heat has to be delivered, and the excess must be recovered and used at low pressure for other purposes. The lower the pressure of the process steam the more power could the steam have generated in an engine or turbine.

On balance it is almost certain that within the limits imposed by heat transfer in the plant, the steam pressure should always be as low as it is possible to arrange it. The steam pipes will be larger and more costly and will take up

more room, but the larger pipe at the lower temperature will probably not lose any more heat than the smaller pipe at the higher temperature as has been discussed in Section 175.

393. FLASH TANK SIZE. Flash steam is liberated from overheated water almost instantaneously. The flashing can therefore be very violent. Any flash vessel must be sufficiently large to allow proper separation of the steam from the water without the carry-over of a lot of water droplets. There are two rules to guide us.

The first is the amount of steam that experience with steam accumulators has shown can safely be liberated from a water surface without undue carry-over. A good conservative figure is

$$\begin{array}{c}\text{Pounds of flash steam per sq. ft.}\\ \text{of water surface per hour}\end{array} = 3 \times \begin{array}{c}\text{absolute pressure}\\ \text{on water}\end{array}$$

This rule gives the flash rates shown in Table LVI :—

TABLE LVI. PERMISSIBLE WEIGHT OF FLASH STEAM FROM WATER SURFACE

(Ruths)

PRESSURE ON WATER SURFACE	LB. FLASH STEAM/HOUR/SQ. FT. WATER SURFACE
Atmos.	44
5 psi.g.	59
10 ,,	74
20 ,,	104
30 ,,	134
40 ,,	164
50 ,,	194
100 ,,	344
150 ,,	494
200 ,,	644

This assumes that all the flash occurs at the water surface in the vessel. In most cases flash tanks are arranged so that the overheated water enters above the water level and drops in a broken flashing stream into the body of the water. This gives a much larger flashing surface than the cross section of the vessel. The 3 × psi.a. rule therefore always gives us the upper limit for the flash tank size.

The other approach is to limit the steam velocity in the flash tank to such a speed that entrainment of water drops is unlikely. This limiting speed can be little better than a guess. If the splashes are in large drops the steam velocity can be much larger than if the splashed drops are very small. Hausbrand (Section 805) gives data from which we can construct Table LVII of steam velocities that will exert on water drops of various sizes pressures equal to their weights.

What are the likely sizes of drops flashed off with the steam ? Perhaps they are of all sizes from ·1 in. downwards. We are probably being fairly safe if we use the first column of Table LVII for drops of ·005 in. diameter for the design of our flash tanks, provided we say that this will give us the lower size limit. We can then select the most convenient size lying between the limits that the two methods of size fixing have given.

If we make the cross section of the tank such that the steam flow is not great enough to carry away any appreciable amount of water there is no need to make the tank deep. Its depth becomes merely a matter of convenience from any other point of view—the things we may want to put inside it, the size of the particular piece of pipe that is lying on the scrap heap that is going to re-live in a new role, etc.

TABLE LVII. STEAM VELOCITIES EXERTING FORCES ON WATER DROPS EQUAL TO THEIR WEIGHTS

(Hausbrand)

STEAM PRESSURE	STEAM VELOCITY IN FT./SEC. ON DROPS OF DIAMETER :—							
	·005-IN.	·01-IN.	·02-IN.	·05-IN.	·1-IN.	·15-IN.	·2-IN.	·25-IN.
25-in. vac.	13·1	22·6	32·0	50·6	72·1	88·2	101·3	110·5
20-in. ,,	9·6	16·6	23·5	37·2	52·9	64·8	74·4	81·2
15-in. ,,	7·9	13·6	19·3	30·5	43·7	53·2	61·1	66·6
10-in. ,,	6·9	11·9	16·9	26·8	38·1	46·6	53·5	58·4
5-in. ,,	6·2	10·7	15·2	24·0	34·2	41·9	48·1	52·5
Atmos.	5·7	9·8	13·9	22·0	31·3	38·3	44·0	48·0
10 psi	3·5	6·0	8·6	13·5	19·2	23·5	27·0	29·5
20 ,,	2·6	4·4	6·2	9·8	14·0	17·1	19·6	21·4
30 ,,	2·0	3·5	4·9	7·8	11·1	13·6	15·6	17·0
40 ,,	1·7	2·9	4·1	6·4	9·1	11·1	12·8	14·0
50 ,,	1·4	2·4	3·5	5·5	7·8	9·6	11·0	12·0

394. COLLECTION OF FLASH FROM HOT-WELL—SIMPLE. Fig. 217 shows the simplest and easiest arrangement for the collection of flash steam from condensate returned from traps to a hot well for boiler feed. The traps from plant working at 100 psi discharge into a tank at atmospheric pressure. Each pound of condensate contains 309 Btu. The flash steam is condensed by the make-up water. Let us see how much make-up at 60° F. can be heated to 212° F., supposing that the amount of condensate is 1,000 lb./hour.

Input .. .. 1,000 lb. Condensate containing 309 Btu/lb.
x lb. Make-up containing 28 Btu/lb.

Output .. Hot water mixture at 212° F. containing 180 Btu/lb.

$$\therefore \quad (1{,}000 \times 309) + (x \times 28) = (1{,}000 + x)\,180$$
$$309{,}000 + 28x = 180{,}000 + 180x$$
$$129{,}000 = 152x$$
$$849 = x = \text{Make-up.}$$

Therefore the minimum amount of make-up to quench the flash is 850 lb./hour. As it is very unlikely that the discharge of condensate is even and free from peaks, or that the draw-off from the tank is smooth, there must necessarily be a much larger addition of make-up than 850 lb./hr. to make sure of dousing all the flash all the time, or much flash steam will be wasted. This arrangement, assuming a large draw-off, is shown in Fig. 217.

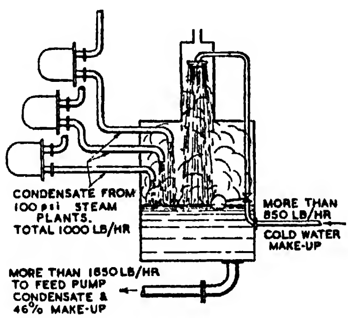


FIG. 217. SIMPLE FLASH COLLECTION IN HOT-WELL

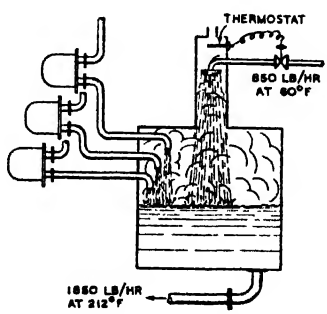


FIG. 218. THERMOSTATIC FLASH COLLECTION IN HOT-WELL

The arrangement shown in Fig. 218 ensures that only the minimum 850 lb. of make-up water is added. The tank in this case must be larger to accommodate peak draw-offs when the condensate input is small. The make-up input, which condenses the flash steam, is controlled by a thermostat above the cold water spray. When steam reaches the thermostat bulb cold water is admitted to the spray until there is just a breath of vapour coming out of the top vent.

This factory must have a very bad condensate return system if the hot-well can take 45 per cent. of make-up, or, more charitably, much of the steam is used for heating by blower. What is to be done if only 10 or 20 per cent. of make-up can be accommodated ?

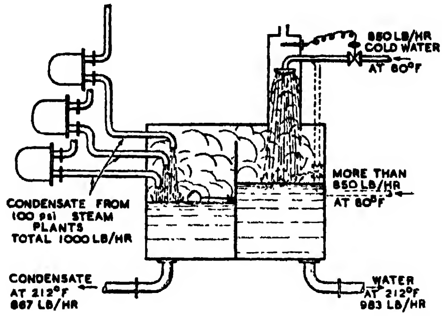


FIG. 219. THERMOSTATIC HOT-WELL FLASH COLLECTION WITH UNCONTAMINATED CONDENSATE

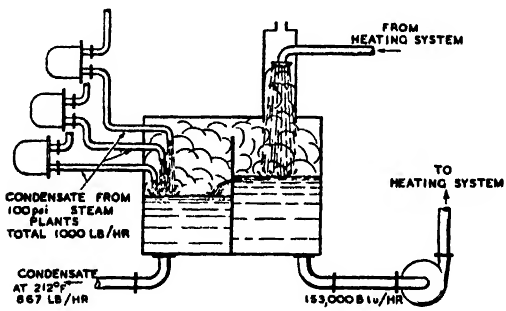


FIG. 220. HOT-WELL FLASH COLLECTION USING FLASH HEAT FOR SPACE HEATING

Figs. 219 and 220 show two ways. The flash steam is led to a separate spray condenser where it heats water that does not mix with the condensate. In Fig. 219 cold water is heated to 212° F. The control is by thermostat and the water temperature is held constant, but the water quantity will fluctuate with the amount of flash evolved. A float control is shown dotted. This would prevent the tank running dry, but might waste flash steam at times of small water demand and large condensate output, while at times of heavy water draw and small condensate flow the water will be much cooler than 212° F.

We can calculate the amount of condensate left in the hot-well and the heat added to the cold water thus :—

1 lb. of condensate at 100 psi contains ..	309 Btu
1 lb. of condensate at Atmos. contains ..	180 Btu
Heat available to produce flash	129 Btu

Amount of flash .. .. $\frac{129}{971}$ = ·133 lb./lb.

1 lb. steam at Atmos. contains .. .. 1,151 Btu
·133 lb. steam at Atmos. contains .. .. .. 153 „

Amount of condensate left .. .. 1 − ·133 = ·867 lb.

The amount of cold water that will be heated will be the same as in Fig. 218, namely .. .. .. 850 lb.

Check :—

133 lb. flash containing 1,151 Btu/lb.
x water containing 28 Btu/lb.

$$(133 \times 1{,}151) + (x \times 28) = (133 + x)\ 180$$

$$129143 = 152x$$

$$850 = x$$

The amount of hot water output will be
850 + 133 = 983 lb. at 212° F.

The amount of condensate is .. .. .. 867 lb. at 212° F.

So that we see that by this means we reduce the water in the hot-well in two ways. We add no make-up and we lose from the hot-well 133 lb. of potential condensate as flash. In Fig. 219 the float in the hot-well draws any necessary make-up from the flash-quenching tank, so that the boiler feed will not be starved. A float control to the spray is shown dotted, in case ample water is more important than water temperature.

In Fig. 220 the flash steam is condensed by the fast circulation of warm water which passes through the office heating system. Let us see how big an office or shop we can heat with this flash. Let us say that we require 5 Btu per cu. ft. of office per hour.

Heat in flash .. .. 133 × 1,151 Btu/hour.

Cu. ft. heatable .. .. $\frac{133 \times 1{,}151}{5}$ = 30,617

or a building 50 ft. × 50 ft. × 12 ft. 3 in.

Fig. 220 only provides a solution in winter when space heating is required.

The amount of water in the heating circuit will increase due to the condensation of the flash steam. The excess will flow over the division plate into the condensate compartment.

395. COLLECTION OF FLASH FROM HOT-WELL—COMPOUND. The arrangements shown in Figs. 217 to 220 collect all the heat in the flash, but it is collected at atmospheric pressure. In many factories there is a surplus of heat at 212° F. and in most factories there is a use for low pressure steam. Let us assume that there is a 10 psi steam main. Fig. 221 shows the arrangement that would be adopted. The traps discharge into a flash tank at 10 psi from

which the flash steam is piped to the 10 psi main. A large low pressure trap passes the condensate from the flash tank to the hot well, where the remaining flash is quenched with make-up. The quantities are :—

Input to flash tank ..		1,000 lb./hr.
containing .		309 Btu/lb.
Sensible heat at 10 psi ..		208 Btu/lb.
Surplus heat for flashing		101 Btu/lb.
Latent heat at 10 psi .		953 Btu/lb.
Amount of 10 psi flash .	$\frac{101}{953}$	·106 lb./lb.
Total heat in 10 psi steam		1,161 Btu/lb.
·106 lb. will carry away		123 Btu
We shall therefore get 106 lb. of flash steam carrying ..		123,000 Btu
The 10 psi condensate will weigh 894 lb. and will hold		186,000 Btu

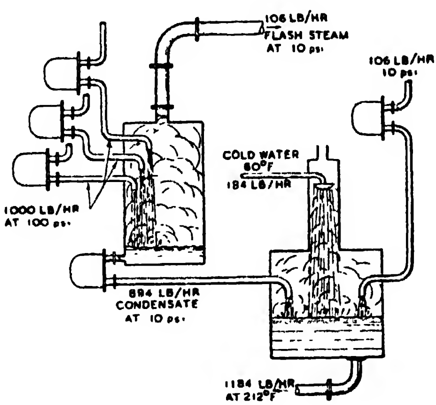


FIG. 221. COMPOUND HOT-WELL FLASH COLLECTION

The 10 psi flashed steam will condense in the plant to which it is piped and its condensate can return to the same hot-well.

In the hot-well there will be an input of

894 lb./hr. of flashed condensate from flash tank
and 106 lb./hr. of flashing condensate from 10 psi users

1,000 lb./hr. of condensate containing 208 Btu/lb.
plus x lb./hr. of make-up containing 28 Btu/lb.

The hot-well output is water at 212° F. containing 180 Btu/lb.

$$(1{,}000 \times 208) + (x \times 28) = (1{,}000 + x)\ 180$$
$$208{,}000 + 28\,x = 180{,}000 + 180x$$
$$28{,}000 = 152x$$
$$184 = x = \text{make-up.}$$

The output from the hot-well will be 1,184 lb., the make-up now being only 15·6 per cent. instead of the 46 per cent. that was needed to quench the flash in Fig. 217.

The size of the flash tank will be :—

By the 3 × psi.a. rule

Each sq. ft. will liberate	74 lb. flash steam/hour
106 lb./hr. will need	1·43 sq. ft.
or a diameter of	1 ft. 4 in.

By the drop weight/velocity rule (*See* Table LVII.)

At 10 psi for ·005 in. drops, the velocity must not exceed	3·5 ft./sec.
Steam at 10 psi has a volume of	16·5 cu. ft./lb.
106 lb. of 10 psi steam has a volume of ..	1749 cu. ft./hour
Flow will be	·486 cu. ft./sec.
Velocity of 3·5 ft./sec. will need an area of	·139 sq. ft.
Diameter of flash tank	$5\frac{1}{16}$ in.

A piece of pipe 9 in. or 12 in. in diameter will probably prove satisfactory.

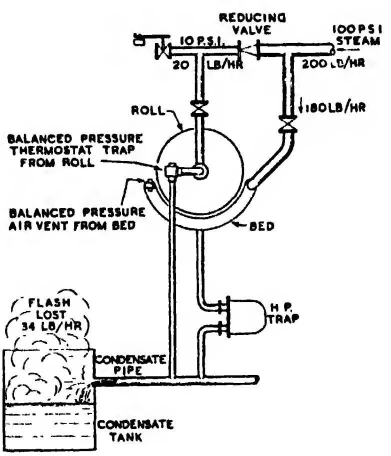


FIG. 222. LAUNDRY CALENDER WITHOUT FLASH COLLECTION

396. LAUNDRY CALENDER. In a Laundry calender or decouden the process is twofold, drying and surface-finishing. The latter calls for a high temperature. The high temperature is applied by means of high pressure

steam to the calender bed, only sufficient heat being supplied to the roll to keep its clothing dry. The roll heating is done by low pressure steam. Fig. 222 shows the normal arrangement and gives an indication of the possible steam consumption and the probable flash loss.

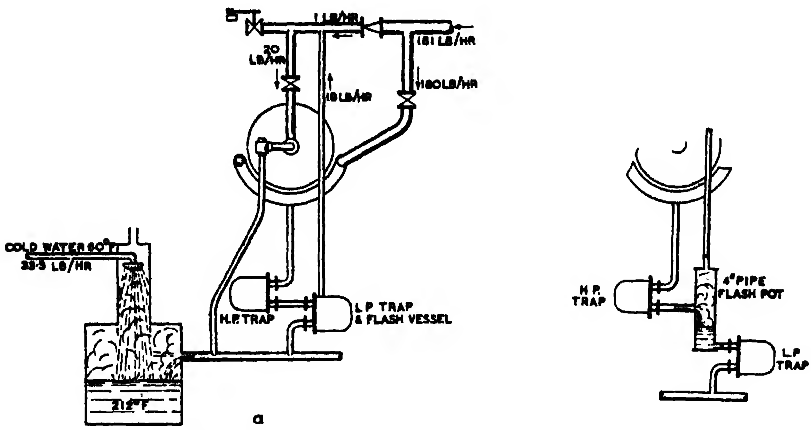


FIG. 223. LAUNDRY CALENDER WITH FLASH COLLECTION FOR HEATING ROLL

In Fig. 223a the condensate from the bed is allowed to flash down to 10 psi in an extra trap, and a steam pipe is fitted to the top of this trap and connected to the low pressure steam supply to the roll. A small reducing valve is fitted to supply any make-up steam that the roll may call for. The diameter of the flash trap or flash pot by the 3 × psi.a. rule is 6¾ in. and by the ·005 in. water drop rule is 2⅛ in. If the trap used as flash pot is 3 in. or 4 in. in diameter this will be quite adequate. If a smaller trap is used then a flash tank should be inserted between the high pressure trap and the low pressure trap as shown in Fig. 223b.

In Fig. 223a, 181 lb. of steam per hour are used to heat the calender and the final flash heats 33·3 lb. of cold water to 212° F. and 181 lb. condensate are recovered. In Fig. 222, 200 lb. of steam per hour are used for the calender only, and an additional 5 lb. of steam would be needed to heat 33·3 lb. of cold water while only 166 lb. of condensate are recovered. So that 181 lb. of 100 psi steam in Fig. 223a are doing the same work as would call for 205 lb. of steam in Fig. 222—a saving of 11·7 per cent.

It is unlikely that a flash collecting system as small as this can possibly be worth while. Anyhow, a reducing valve passing 1lb./hr. is quite impractical. In such cases the condensate from a number of plants should be collected into one larger flash tank, when the quantities of flash steam may warrant the complication and expense.

397. FLASH FROM PAPER MACHINES. The collection of flash steam from paper machines is a regular part of their technique. It is a compound system, generally in three stages. Where the web enters the machine it is really a sheet of water held together by a loose felt of fibres. Where it leaves

the machine it is compressed sheet of fibre containing hardly any water. At the dry end of the machine drying is more difficult due to the low water content, so that a higher temperature is needed than at the wet end. The higher temperature at the dry end also imparts finish to the paper. At the wet end heat transfer is much easier so that a lower steam pressure can be used. In machines

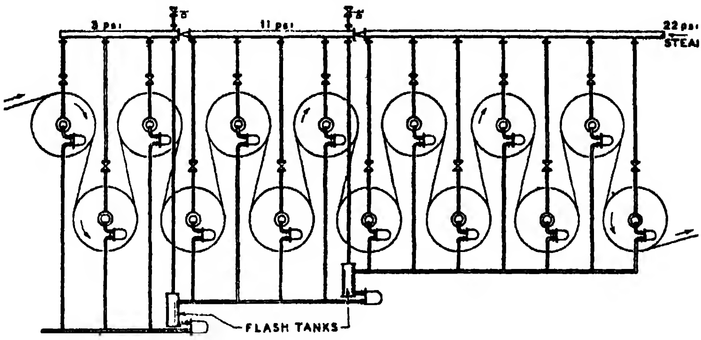


FIG. 224. FLASH COLLECTION IN STAGES ON 14-CYLINDER PAPER MACHINE

with a fair number of cylinders it is usual to use three steam pressures on the machine. The steam flow is in the opposite direction to the paper flow. Relatively high pressure steam is used at the dry end on a large number of cylinders, the steam pressure is reduced for a lesser number of intermediate cylinders, and a still lower pressure is used for relatively few wet end cylinders. Each cylinder should be individually trapped and the condensate from each group of traps should go to a flash tank from which the flash steam is piped to the next lower pressure. See Fig. 224.

398. **HIGH PRESSURE HEATING WITH LOW PRESSURE EFFICIENCY.** Suppose we have a battery of 20 Unit heaters, Fig. 225, taking steam at 60 psi and each liberating 100,000 Btu per hour. If the trapping system is good the condensate will quickly be got rid of and will only lose, say, 10° F. on its way to the hot-well, where any surplus condensate heat is lost by flash.

The total heat liberated will be 2,000,000 Btu of latent heat per hour plus 10 Btu of sensible heat per lb. of condensate.

Latent heat of 60 psi steam	905 Btu/lb.
Steam consumption will be $\frac{2,000,000}{905}$ ..	2,210 lb./hour
Sensible heat lost by condensate	22,100 Btu/hour
Sensible heat of condensate 277—10 ..	267 Btu/lb.
Sensible heat in condensate at atmos. pressure	180 Btu/lb.
Heat available for flashing	87 Btu/lb.
Amount of flash at Atmos. Pressure $\frac{2,210 \times 87}{971}$ =	198 lb./hr.

The flash is 9 per cent. of the total input steam.

Now suppose we only use 18 heaters on live steam, collect the condensate from these in a flash tank and lead the flash steam into the two remaining heaters. Owing to the lower heat transfer at 2 psi, which is the pressure that might be chosen for the flash tank, it will be necessary to raise the temperature and pressure of the steam on the 18 live steam heaters to compensate. We can make a rough estimate thus :—

The air in the shop is taken at	62° F.
The temperature difference at 60 psi is 307 − 62 ..	245° F.
The temperature difference at 2 psi is 219 − 62 ..	157° F.

The heaters supplied with flash steam will have a lower heat transfer in ratio, say, 245 to 157.

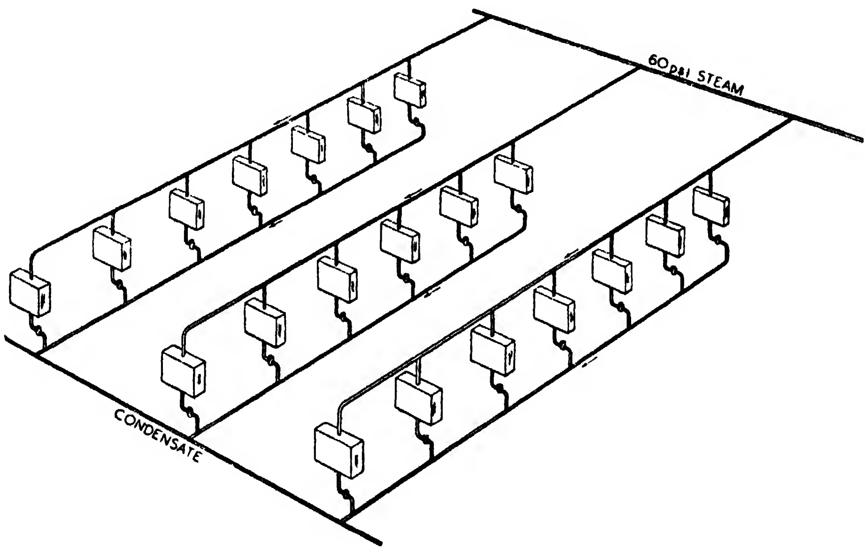


FIG. 225. HIGH PRESSURE UNIT HEATER SYSTEM

Two heaters should condense about $\frac{2210}{20} \times 2 \times \frac{157}{245} = 141$ lb./hour

Two heaters will therefore be insufficient to condense the flash steam, so we must use three.

The flash heat to be transferred is $198 \times 966 = 191{,}268$ Btu.

Three heaters at 2 psi will transfer $\frac{100{,}000 \times 157 \times 3}{245} = 192{,}245$ Btu which looks as if it will balance out all right.

We shall then have roughly $2{,}000{,}000 - 190{,}000 = 1{,}810{,}000$ Btu to be transferred by 17 heaters or 106,500 per heater.

The temperature difference must therefore be raised in ratio 100 to 106·5.

The temperature difference must be $245 \times \frac{106 \cdot 5}{100} = 261°$ F.

The steam temperature must be $261 + 62 = 323°$ F. which is the saturation temperature of steam at 79 psi.g. or say 80 psi whose latent heat is 893.

The steam consumption will be $\frac{1{,}810{,}000}{893} = 2{,}027$ lb./hr. at 80 psi.

The sensible heat in 80 psi condensate is 295, with, say, 11 Btu lost by radiation.

The sensible heat in 2 psi condensate is 187, with a latent heat of 966.

The amount of flash will be $\frac{(295 - 11 - 187) \times 2{,}027}{966} = 204$ lb./hr.

The flash heaters will give up $204 \times 966 = 197{,}064$ Btu

The condensate from these heaters contains 187 Btu/lb., from which we can assume a loss of 7° F.

The total heating done by 2,027 lb. of 80 psi steam will be :—

Latent heat from 80 psi steam is $2{,}027 \times 893$ ..	1,810,111 Btu
Sensible heat radiated from 80 psi condensate is $2{,}027 \times 11$	22,297 Btu
Latent heat in flash steam at 2 psi is 204×966 ..	197,064 Btu
Sensible heat radiated from 2 psi condensate is 204×7	1,428 Btu
Total heating from 2,025 lb./hr. of 80 psi steam	2,030,900 Btu

The total heating done by the original 2,210 lb. of 60 psi steam was :—

Latent heat from 60 psi steam $2{,}210 \times 905$ ..	2,000,050 Btu
Sensible heat radiated from condensate $2{,}210 \times 10$	22,100 Btu
	2,022,150 Btu

The new system is delivering slightly too much heat. Its steam can be reduced to :—

$$\frac{2{,}027 \times 2{,}022}{2{,}031} = 2{,}018 \text{ lb. of 80 psi steam/hour.}$$

The steam saving is $2{,}210 - 2{,}018 = 192$ lb./hour.

This is 8·6 per cent., but the total heat in 80 psi steam is 0·4 higher than in 60 psi steam so we can only rely on getting a saving of about 8·2 per cent. Let us say 8 per cent. to be on the safe side.

Let us make the following assumptions :—

Boiler plant efficiency	68 per cent.
Transmission efficiency	90 per cent.
Calorific value of coal	12,000 Btu/lb.
Hours per week—one shift	50
Heating weeks per year	30

The annual coal consumption on the original system was :—

$$\frac{2{,}210 \times (1{,}182 - 170) \times 50 \times 30}{2{,}240 \times 12{,}000 \times \cdot 68 \times \cdot 9} = 204 \text{ tons coal/year.}$$

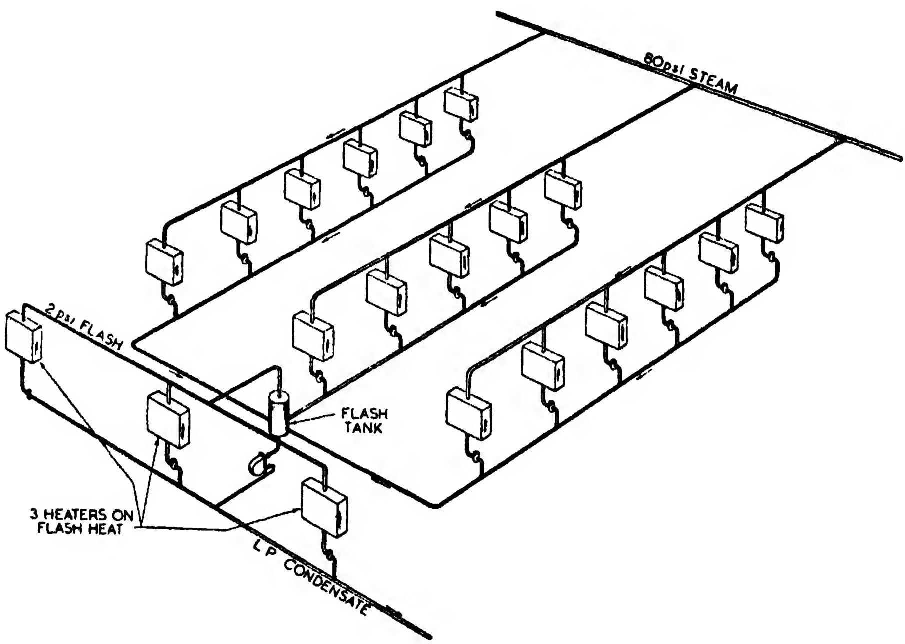


FIG. 226. HIGH PRESSURE UNIT HEATER SYSTEM WITH LOW PRESSURE EFFICIENCY

Assuming a saving of 8 per cent. by the alterations, the coal would be reduced by about 16½ tons a year. On three shifts the saving would be treble this figure. The cost of the alteration should be about £100, as all that is needed is one low pressure trap and a little piping, as shown in Fig. 226.

Now what we have done by using the flash steam is to make the system on high pressure as efficient as a low pressure system working at 2 psi. instead of 60 psi, but there is a little more to it than this. Suppose we were to instal a system working all over at 2 psi ; what would this entail ?

We wish to transfer 2,000,000 Btu/hour. We have seen that at 2 psi three heaters will transfer about 200,000 Btu/hour, so that instead of 20 heaters on 60 psi, or 80 psi + 2 psi, we should need 30 heaters if all were on 2 psi.

In practice the siting and number of heaters are probably decided by considerations of heat distribution in the building, and instead of using 30 heaters in place of 20 we might use 20 heaters each 50 per cent. larger. This would add considerably to the cost because all the piping would have to be larger in size. Just what would the extra piping amount to ?

With a liberation of 2,000,000 Btu/hour we will assume that half the heat is heating the air in the building that must be passed through for ventilation, and that the other half is lost through the fabric of the building.

If the ventilation heating is such as to require 2 Btu/cu. ft. of building per hour (Table LXVII, Chapter 20) it means that the building to be heated has a content of 500,000 cu. ft.

If it is 20 feet high the floor area must be 25,000 sq. ft.

Let us assume that it is 100 feet wide by 250 feet long.

The longest steam pipe will be something less than 250 feet long.

At 60 psi we can tolerate a pressure drop of about 2 psi/100 ft.

The volume of 60 psi steam is 5·84 cu. ft./lb.

With three rows of heaters each steam pipe will have to handle about 740 lb./hour or 12·3 lb./min.

Reference to Fig. 49 in Section 174 tells us that we must use a 2 in. pipe, and Fig. 50 tells us that the steam velocity in this pipe will only be 50 ft./sec.

At 2 psi we can afford to lose about 1 psi in the whole pipe length, or ·4 psi/100 ft.

The steam to be passed at the lower pressure is only 11·3 lb./min.

The volume of 2 psi steam is 23·8 cu. ft./lb.

Figs 49 and 50 tell us that we shall need a 3 in. pipe.

Table XXXI tells us that 3 in. pipe will cost 33 per cent. more than 2 in. pipe and we need 50 per cent. more heaters, so we probably will not be far wrong if we say that the 2 psi system will cost about 40 to 45 per cent. more than the 60 psi system.

Steam at 2 psi has the following advantages : The steam could have generated some 30 kW in expansion in an engine from 60 psi to 2 psi ; owing to the greater number of heaters and their lower temperature, the heating will be much kinder and less inclined to give hot and cold spots, which is one of the disadvantages of unit heating. These advantages must be set against the extra cost. It almost always pays to do the right thing, although it may not pay to convert the wrong thing to the right thing at a later date—because two lots of money have to be spent. If, for some reason, it is decided to use high pressuer then the pressure should be even higher than intended and should be used on 17 heaters with the flash going to the other three heaters. We can be fairly categorical in making recommendations. A new plant should be put in for low pressure. A high pressure plant where the flash is lost, or is troublesome to collect, should be put on to higher pressure and about 15 per cent. of the heaters should take flash.

Of course, all the foregoing is not confined to unit heaters. There are many heating systems to which it can be applied. Any long-tube or multi-bank heater can be cut into sections and a small section used for flash steam, while the pressure on the rest is raised to compensate.

Actually the unit heater problem is given here as an example pure and simple. There is a far better and simpler way of getting low pressure efficiency out of a high pressure unit heater system. If the condensate pipes are left unlagged and in fact if some sections are replaced with gilled pipes all the excess sensible heat can be taken out of the condensate without any need for flash. If the gilled piping is used fairly lavishly for the condensate return the condensate can be cooled down to any desired figure, thus enabling the economiser (if one is fitted to the boiler) to take more heat out of the flue gases, thereby making a double saving. Had this cat been let out of the bag at the beginning of this section, the reader would have lost interest in a very nice little bit of flash technique.

There is another way. If the unit heaters are drained by expansion traps, the traps can be set to operate at 212° F. and there will be no flash from such condensate. This cannot be done without raising the steam pressure, because part of each unit heater will be waterlogged with low temperature water and the heat transfer will be lowered in the bottom part of each heater.

399. COLLECTING THE HEAT FROM BOILER BLOWDOWN. Blowdown contains a lot of heat. Blowdown is a scale-producing and a sludge-bearing liquid and may give trouble on the surfaces of heat exchangers. Heat exchange through a surface is always more expensive than exchange, or rather marriage, by direct contact. Direct contact, however, does limit the heating to something that can tolerate, chemically, physically or economically the addition of water.

Continuous blowdown is much more convenient from a heat recovery point of view than intermittent blowdown. It is, however, not so fool-proof and the tiny blowdown cocks sometimes get blocked or wear badly. It may, in some plant, be inconvenient to lose any boiler power by blowing down during peak loads.

A useful compromise can sometimes be made by using continuous blowdown for certain convenient periods—say during the whole afternoon, or over the whole night shift. Provided the period runs into hours the heat can usually be collected.

Take two examples to see what can be done with the blowdown heat :—

(*a*) A Lancashire boiler evaporating 7,000 lb./hr. at 150 psi.g with 5 per cent. blowdown ; feed water at 150° F. ; no superheater ; boiler efficiency 68 per cent. by steam meter ; process steam used at 20 psi.g.

(*b*) Water-tube boiler evaporating 100,000 lb./hr. at 650 psi.g with 1 per cent. blowdown ; feed water at 210° F. ; superheat at 800° F. ; boiler efficiency 82 per cent. by steam meter ; process steam used at 70 psi.g and at 10 psi.g.

(*a*) The total heat in 150 psi saturated steam is 1,197 Btu and the heat in the feed is 118. The boiler therefore has to supply 1,197 − 118 = 1,079 Btu/lb.

The total heat addition in the boiler is 7,000 × 1,079 Btu.

With 12,000 Btu coal and 68 per cent. boiler efficiency the coal consumption is :—

$$\frac{7{,}000 \times 1{,}079}{12{,}000 \times \cdot 68} = 926 \text{ lb./hr.}$$

The sensible heat in boiling water at 150 psi is 339 Btu.

1 lb. of blowdown has needed 339 − 118 = 221 Btu.

A blowdown of 5 per cent. is 350 lb./hr. requiring

$$221 \times 350 = 77{,}350 \text{ Btu.}$$

So that the real heating done by the boiler is

$$(7{,}000 \times 1{,}079) + (221 \times 350) = 7{,}630{,}350 \text{ Btu.}$$

The blowdown therefore is just about 1 per cent. of the heat put into the boiler.

The coal wasted by the blowdown, if it all goes to drain, is
9¼ lb./hr.
or 10 tons/year on 1 shift
or 30 tons/year on 3 shifts.

If there is plenty of cold water to be heated, a good countercurrent heat exchanger can save almost the whole of the loss. If there is already plenty of waste heat used for water heating so that water heating offers less scope, matters are not quite so simple.

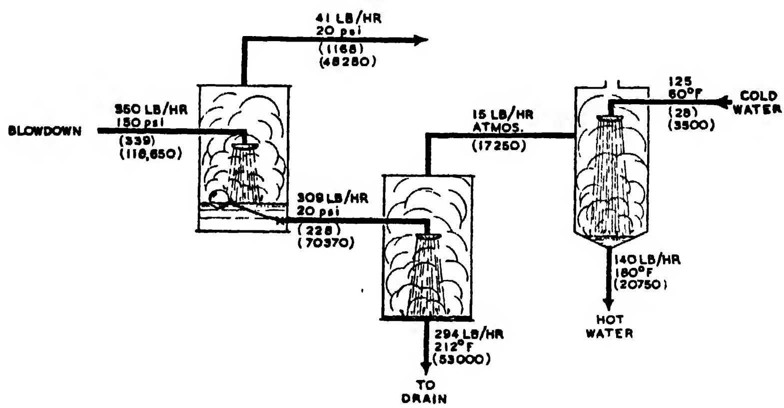


FIG. 227. TWO STAGE BLOW-DOWN FLASH COLLECTION

Fig. 227 shows an arrangement with the simplest possible plant with no heat exchanger. The blowdown is sprayed into a flash tank whose steam outlet is connected to the 20 psi process main.

The blowdown water contains	339 Btu/lb.
Boiling water at 20 psi contains	228 Btu/lb.
Surplus heat	111 Btu/lb.
Latent heat at 20 psi is	940 Btu/lb.
Amount of flash steam will be $\dfrac{350 \times 111}{940}$ =	41·3 lb./hr.
Total heat in 20 psi steam is	1,168 Btu/lb.
So that the total heat carried away by flash is ..	48,238 Btu/hr.
The partly cooled blowdown now consists of $350 - 41$ =	309 lb./hr.
and contains $(350 \times 339) - 48{,}238$ =	70,412 Btu/hr.

The partly cooled blowdown is passed to a second flash tank at atmospheric pressure, where it can only hold 180 Btu.

There is therefore a heat surplus of $228 - 180 = 48$ Btu/lb.

This will give a flash of $\dfrac{48}{971} = \cdot 0495$ lb./lb. of blowdown.

From 309 lb. of blowdown there will be a flash of 15·3 lb.

containing say $15 \times 1{,}151 = 17{,}265$ Btu.

This leaves $309 - 15 = 294$ lb./hr. containing 53,147 Btu to go to drain.

The flash at atmospheric pressure goes into a spray condenser where it heats 125 b. of water per hour from 60° F. to 180° F.

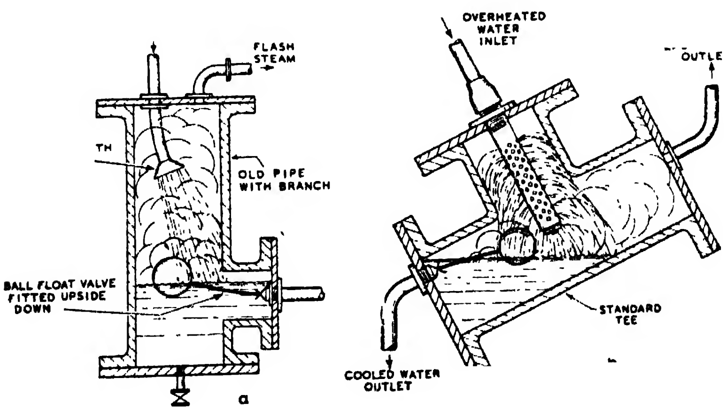


FIG. 228. FLASH TANKS MADE FROM SCRAP TEES

If the sizes of the flash tanks are worked out according to the rules given in Section 388 it will be found that 12-in. pipes will suffice amply for the flash tanks. A 12-in. pipe will also do for the spray condenser. So that 3 old short

lengths of 12-in. pipes or tees, plus three shower bath sprays coupled together with a minimum of piping will save, for a generation, 6 to 20 tons of coal a year.

The only snag would seem to be, how to fit a ball float valve into a 12 in. pipe. Fig. 228 shows two cheap simple ways of making flash tanks from piping lengths.

It may not be necessary, possibly not even desirable, to break the condensate up through the sprays shown in Fig. 228. In the author's factory there are flash vessels working with sprays and with open ended pipes. Both types seem to work equally well.

If the condensate pipes leading to the flash pot are large, then much of the flashing will have taken place in the pipes and open ended inlets to the flash tank will probably be best, the flash tank acting then primarily as separator.

If the condensate pipes are small and heavily loaded, little flash will occur in the pipes and the inlets to the flash tank should probably be fitted with sprays.

400. HIGH PRESSURE BLOWDOWN. Let us now look at case (*b*) where a boiler at 650 psi is evaporating 100,000 lb./hr. with 1 per cent. blowdown ; steam superheated to 800° F. ; feed at 210° F. ; process steam at 70 psi and 10 psi.

Total heat in 650 psi 800° F. steam is	1,405 Btu/lb.
Heat in feed 210 − 32 .. .. .. =	178 Btu/lb.
Heat required for each lb. of evaporation 1,405 − 178 .. .. =	1,227 Btu/lb.
Net work done by boiler	1,227 × 100,000 Btu/hr.
Coal needed at 82 per cent. efficiency $\frac{1,227 \times 100,000}{12,000 \times \cdot 82}$ .. .. =	12,480 lb./hr.
Sensible heat at 650 psi is	485 Btu/lb.
Blowdown heat at 1 per cent. is (485 − 178) 1,000 =	307,000 Btu/lb.
That is $\frac{307,000 \times 100}{(1,227 \times 100,000) + 307,000}$ =	·25 per cent.
Blowdown coal consumption is ·25 per cent. of 12,470 lb./hr.	31 lb. coal/hr.
On a three shift week this means	100 tons coal/year.

A heat exchanger at 650 psi is an expensive piece of plant and anyhow is seldom applicable as there may not be sufficient cool water to heat. The best heat exchanger system would be to take as much flash heat out of the blowdown as possible first and to heat exchange the rest.

Let us see what we can do by flash alone. Fig. 229 shows an arrangement. There is no need to plod through the arithmetic ; previous examples have shown how it is done. The result is remarkable. We only need some lengths of 12 in. pipe, two float valves and a little piping and we save 74 per cent. of the blowdown heat or 80 tons of coal a year.

By more elaborate methods we can secure almost the whole of the heat; we can get more low pressure steam; we can heat nearly twice the water; and can, on the top of all this get 1,760 lb./hour of distilled water. This more highfalutin arrangement is described in Chapter 17, Section 496.

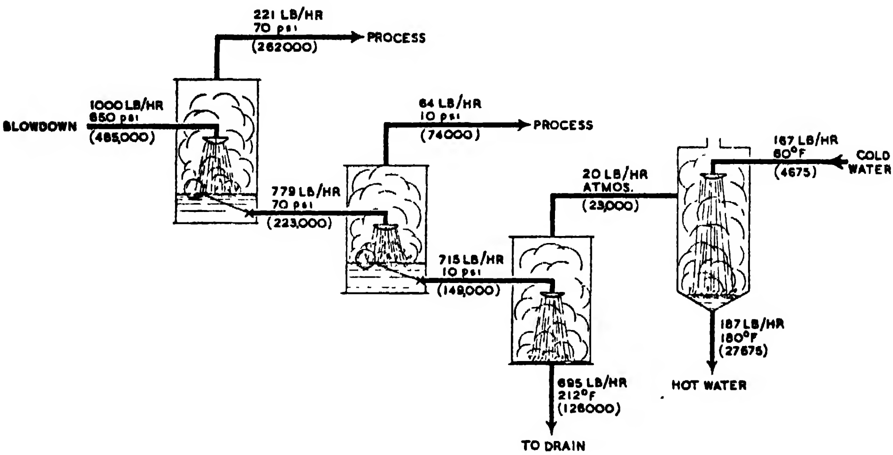


FIG. 229. THREE STAGE FLASH COLLECTION FROM HIGH PRESSURE BLOWDOWN

Of course continuous blowdown is almost a *sine qua non* for these heat recovery systems. If flash collection were to be carried out with blowdown done once daily the plant might have to have 50 times the capacity; the low pressure steam and hot water would only be produced for a few minutes a day. Continuous blowdown is desirable in itself from a purely boiler point of view. The maximum concentration of the solids in the water is the average concentration, whereas with intermittent blowdown the maximum concentration can be greatly in excess of the average. The only disadvantage of continuous blowdown is that it requires careful watching and the blowdown valves need constant and meticulous care and maintenance.

401. FLASH COOLING—SIMPLE. When condensate, or any other water solution, is allowed to flash by being put under a reduced pressure, the latent heat necessary to evaporate the flash cools the liquid, and cools it instantly and in the cheapest possible plant. This can be used to great effect in many processes where the liquid that needs cooling is to be subsequently concentrated.

For example, sugar solutions deteriorate at high temperatures. On the other hand, due to their high viscosity, it is very difficult to filter them at low temperatures. In any sugar refinery or factory any sugar solution that needs filtering is always subsequently concentrated by evaporation.

Flash cooling is an effective way of cooling instantly, and simultaneously getting a small amount of concentration.

Fig. 230 shows a flash cooler that operated for many years in the author's factory. A 68 per cent. sugar solution has been filtered at 185° F. and immediately after filtration is to be cooled to 160° F. It is sprayed into a vessel maintained at 22 in. vacuum, which is the vapour pressure of a 68 per cent. sugar solution at 160° F. The total heat of a sugar solution of 68 per cent. at 185° F. is 102 Btu/lb. and at 160° F. is 85 Btu. There is therefore a surplus of 17 Btu/lb. to effect evaporation. At 22 in. vacuum the latent heat of water is 1,007 so that ·0169 lb. of water will be evaporated, leaving ·68 of sugar dissolved in ·3031 of water. Bringing this back to percentages the liquor after flashing will have a concentration of 69·2 per cent.

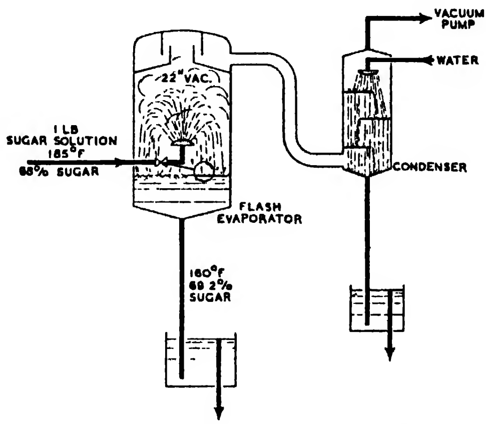


FIG. 230. FLASH COOLING—SIMPLE

By using a high vacuum, really considerable cooling can be done—the so called vacuum refrigeration. The ejector for achieving a very high vacuum calls for a good deal of steam which is exhausted at such a low temperature that its heat must usually be wasted. There are occasional economic applications for vacuum refrigeration, but they are rare.

402. FLASH COOLING—COMPOUND. A greater concentration can be got by doing the flash in two stages and by carrying it further. Fig. 231 shows the modification that was made to the plant in Fig. 230. The sugar liquor enters the first flash vessel and is cooled down considerably but not to final temperature. The flash steam from this vessel goes to a surface condenser or heat exchanger. The liquor is then flashed again to a temperature below the required temperature. The overcooled liquor then passes through the surface condenser where it picks up the heat from the first flash and is brought back to the required temperature. This process is sadly spoilt by the boiling point elevation of sugar solutions which limits the overcooling as it prevents the liquid from being brought back to as high a temperature as the liquor boiling point.

The liquor density has been increased from 68 per cent. to 70 per cent. although in the example it has been assumed that very imperfect (50 per cent.

only) operation of the surface condenser takes place. It may be asked whether this small concentration is worth all the trouble. The only difficulty is the surface condenser, which with small temperature difference and high viscosity liquor must be large and costly. The rest is just two empty vessels and some piping and a jet condenser. What does the saving in evaporation amount to ?

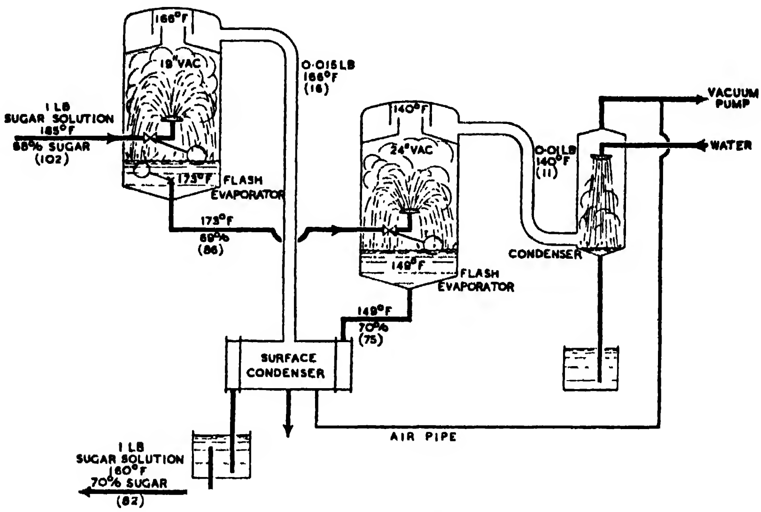


FIG. 231. FLASH COOLING—COMPOUND

A 68 per cent. sugar solution consists of	..	68 sugar 32 water
A 69·2 per cent. sugar solution consists of	..	69·2 sugar 30·8 water
A 70 per cent. sugar solution consists of	..	70 sugar 30 water

For a throughput of 8,500 tons of sugar per week

A 68 per cent. sugar solution consists of	..	8,500 tons sugar 4,000 tons water
A 69·2 per cent. sugar solution contains	..	3,783 tons water
A 70 per cent. sugar solution contains ..	..	3,643 tons water

The simple plant, Fig. 230, saves 217 tons evaporation per week.

The compound plant, Fig. 231, saves 357 tons evaporation per week.

If the evaporation process is 90 per cent. efficient the annual saving is :—

	1939	1957
Cost of steam—per ton	3*s*. 9*d*.	17*s*. 0*d*.
Saving with Fig. 230 plant	£2,000	£9,100
Saving with Fig. 231 plant	£3,300	£15,100

The plant in Fig. 230 is cheap and simple. The compound plant in Fig. 231 is more difficult to control and is expensive. While the compound plant worked fairly well the gain was not always obtained and the saving over the simple plant was not always sufficient to justify the cost and complication in 1939. At 1957 prices matters would have been different, but in the meantime the plant was modified to retain some of the virtue of compound working but doing the secondary heating to water instead of liquor. There may well be factories where the straight compound plant would fit in well.

403. THE SPRAY CONDENSER. In a number of the foregoing examples heat has been recovered from flash steam in a spray condenser. This piece of apparatus is easily the cheapest piece of heat saving plant that exists. It can be made very small and out of almost anything. It is of course just a crude countercurrent jet condenser. It can collect the heat in any kind of waste vapour and there need be no temperature drop in it. The only cause of there being temperatures below 212° F. in it is due to the partial pressure of any air that the vapour may contain.

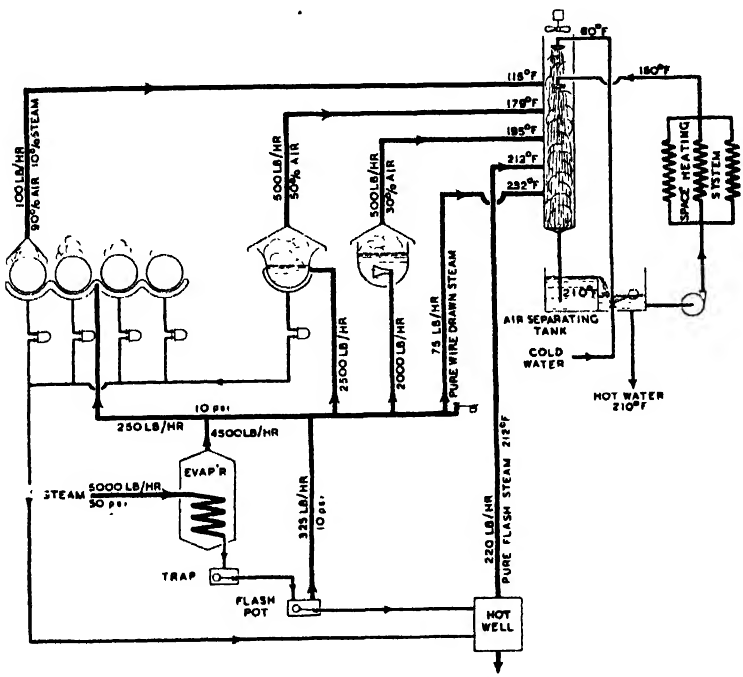


FIG. 232. THE COLLECTION OF LOW GRADE WASTE VAPOUR IN A SPRAY CONDENSER

Suppose we have a plant like that shown in Fig. 232. This is an imaginary factory making no known product. Its main piece of plant is an evaporator which takes 5,000 lb. of 50 psi steam per hour. The evaporator works under a pressure of 10 psi and the vapour from it supplies all the rest of the plant.

A 4-roll calender takes 250 lb./hr., 2,500 lb./hr. goes to a battery of hemispherical pans, 2,000 lb./hr. goes to blowers in open vats. By putting hoods over the wet end of the calender and over the pans and vats much vapour—1,100 lb./hr.—is collected and led to a long spray condenser. The excess 10 psi steam from the evaporator, 75 lb./hr., goes, via the safety valve, to the spray condenser. The flash from the evaporator condensate is taken in two stages, first from a flash pot into the 10 psi main and finally from the hot-well, together with the flash from the 10 psi condensate, into the spray condenser.

404. EFFECT OF AIR IN CONDENSER. The hood over the calender in Fig. 232 must of necessity allow a large air leakage and it is assumed that in the vapour from this plant there are 9 volumes of air to one of steam. Now the air supplies no heat. The heat is all provided by the steam, but the partial pressure of the steam is only 1/10 of 14·7 or 1·47 psi.a. Steam at this pressure only has a temperature of 115° F.—see Section 318, Chapter 10. Similarly the vapour from the pans is assumed to have 50 per cent. of air and only has a temperature of 179° F., while the vapour from the vats is assumed to have 30 per cent. of air and to have a temperature of 195° F.

Table LVIII gives the temperatures of mixtures of air and steam by volume at various pressures.

TABLE LVIII. TEMPERATURE OF STEAM ADULTERATED WITH AIR

STEAM PRESSURE	TEMPERATURES °F OF SATURATED STEAM MIXED WITH AIR OF PER CENT. BY VOLUME										
	0%	5%	10%	20%	30%	40%	50%	60%	70%	80%	90%
25-in. vac.	134	132	130	125	120	115	109	101	92	79	59
20-in. ,,	161	159	157	152	147	141	134	125	115	101	79
15-in. ,,	179	177	174	169	163	157	150	141	130	115	92
10-in. ,,	192	190	187	182	176	169	161	152	141	125	101
5-in. ,,	203	201	198	192	186	179	171	161	150	134	109
Atmos.	212	209	207	201	195	187	179	169	157	141	115
10 psi.g	239	237	234	227	220	212	203	193	179	162	134
20 ,,	259	256	253	246	239	230	221	209	195	177	147
30 ,,	274	271	268	261	253	244	234	222	207	188	158
40 ,,	287	283	280	273	264	255	245	233	218	198	166
50 ,,	298	294	291	283	275	265	255	242	226	206	173
60 ,,	307	304	300	293	284	274	263	250	234	213	180
70 ,,	316	313	309	300	292	282	271	257	241	219	186
80 ,,	324	320	316	308	299	289	278	264	247	225	191
90 ,,	331	328	324	315	306	296	284	270	253	231	195
100 ,,	338	334	330	322	312	302	290	276	258	235	200

Now most of the water passing through the condenser in Fig. 232 is the return water from the heating system and this water is at 150° F. This water could not take up any heat from the calender vapour at 115° F. So the various

vapours must be piped to the condenser in their proper order of temperature with the coolest nearest the top. And, if more than one water supply is sprayed in, the coolest water must go in nearest the top. In this way, as the water gets hotter it meets progressively hotter vapour, and the coolest vapour is certain of condensation as it meets the coldest water.

In Fig. 232 the cold supply is controlled from the tank at the bottom of the condenser. As there is much air in the condenser a large tank is provided at the bottom of the condenser so as to give any entangled air a chance of separating out.

In order to clear the air from the condenser and to provide a gentle draught in the ducts from the plant, a small fan is fitted at the top of the condenser.

When water is heated in a spray condenser such as has just been discussed, it will be saturated with air and will be liable to corrode steel pipes. It may pay to use steel pipes and to renew them every few years. It is probably better, however, to use copper or cast iron.

405. SIZE OF SPRAY CONDENSERS. The cross-section of a spray condenser depends on two things. It must be large enough to prevent any throttling and consequent pressure drop ; the velocity of the vapour must not be sufficient to carry the water drops up too far. There is no chance of the water drops being carried out of the top. Even if the velocity is so high at the bottom as to carry up the drops, condensation is very quick and begins at once. The amount of vapour is therefore continually being reduced and the velocity is continually slowing down. At the bottom of the condenser where the steam first meets the water, the drops have already fallen a considerable distance and are therefore dropping quite fast, and their momentum will prevent their being picked up. It is probably quite safe to use velocities of twice those given in Table LVII. But what size are the water drops ? Probably most of them are ·1 in. or over ; but this is only a guess. Anyhow if there are a few that are smaller they will only be carried up a short way and as the vapour velocity drops due to condensation they will quickly fall, especially as the drops are all the time getting bigger by mopping up vapour.

The height of a condenser must be sufficient to give the water time to take up the heat of the vapour. This is a very quick process. Lengthening the condenser does not proportionally lengthen the time of stay of the drops. Doubling the height adds 40 per cent. to the time the drops take to fall. Speaking very roughly four times the diameter is probably generous ; three times the diameter is probably enough in most cases and twice may often be sufficient. Of course a condenser like that in Fig. 232 is another matter. Probably about two diameters above the last vapour inlet will be all right unless the last inlet carries a lot of vapour or the water from the top spray is very warm.

406. SCALING OF SPRAY CONDENSERS. If the water that is heated into spray condenser contains much temporary hardness and if the temperature is fairly high a great deal of scale may form inside the condenser. Unless provision is made for the removal of this scale it will eventually turn the vessel into a great marble column.

There are several ways of dealing with scale. The first is the provision of plenty of manholes not more than two or three feet apart and on different sides of the condenser. Chains can be hung inside down which the water can fall and the scale will form on the chains which must be frequently drawn and the scale flogged off. If the chains are left too long they will be cemented together and become immovable. The condenser can be make of old junk and can be discarded if the scale gets too thick. (One dairy has made a spray condenser for collecting churn sterilising vapour out of a couple of old oil drums with the ends knocked out.)

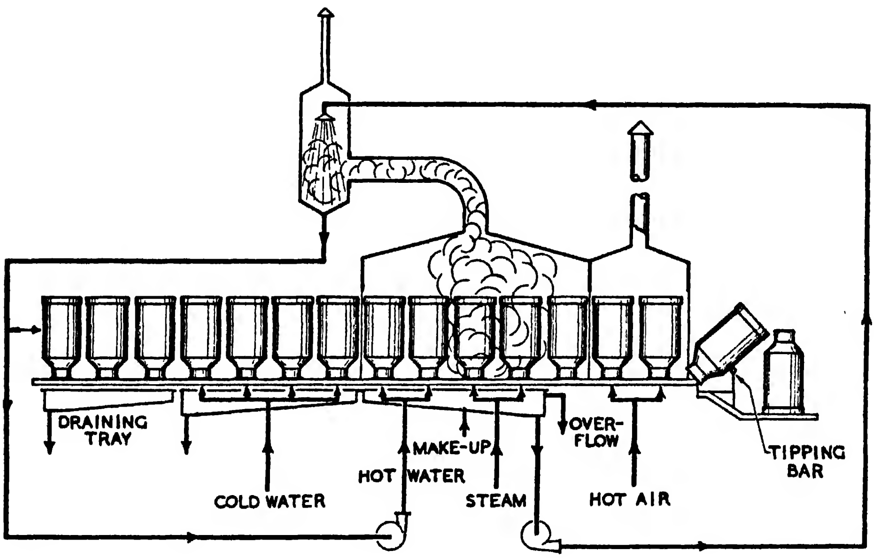


FIG. 233. COLLECTION OF CHURN STERILISING VAPOUR WITH SPRAY CONDENSER

407. CONSTRUCTION MATERIALS FOR SPRAY CONDENSERS AND VAPOUR DUCTS. A spray condenser often collects heat from vapour that is heavily adulterated with air. As the air must have been heated up by the steam to the temperature corresponding to the steam's partial pressure, the vapour will be very wet and condensation will occur fairly heavily in the ducts. The condenser and the ducts leading the vapour to it are liable to severe corrosion. The ducts should be made really good—say of copper, or should be made really cheap, of thin sheet metal tarred internally, and replaced every two or three years. Ducts should be made with a good fall so as to discourage condensate from hanging about. For a permanent condenser there is little to beat cast iron. It is cheap and long-lasting and requires no maintenance. Wood is an excellent material for ducts and for the condenser. It will outlive most sheet iron, and it is light and self-lagging.

408. APPLICATIONS OF SPRAY CONDENSERS. A number of applications have already been mentioned in the examples. Here is

a short list of vapours whose heat can be effectively collected in spray condensers :—

Low pressure flash.
Vapour from vats.
Vapour from pans and evaporators.
Turbine gland leaks.
Exhaust from ejectors.
Waste vapour from open sterilizers.
Blow-off from autoclaves, sterilizers, steamers and pans.
Plant and steam pipe air vents.
Coppers.

Fig. 233 shows the application of a spray condenser to a churn steriliser. The waste vapour goes a long way towards heating the churn washing water. Fig. 234 shows a spray condenser applied to a brewing copper. This heats the water needed for barrel washing, and in some breweries heats the mashing liquor.

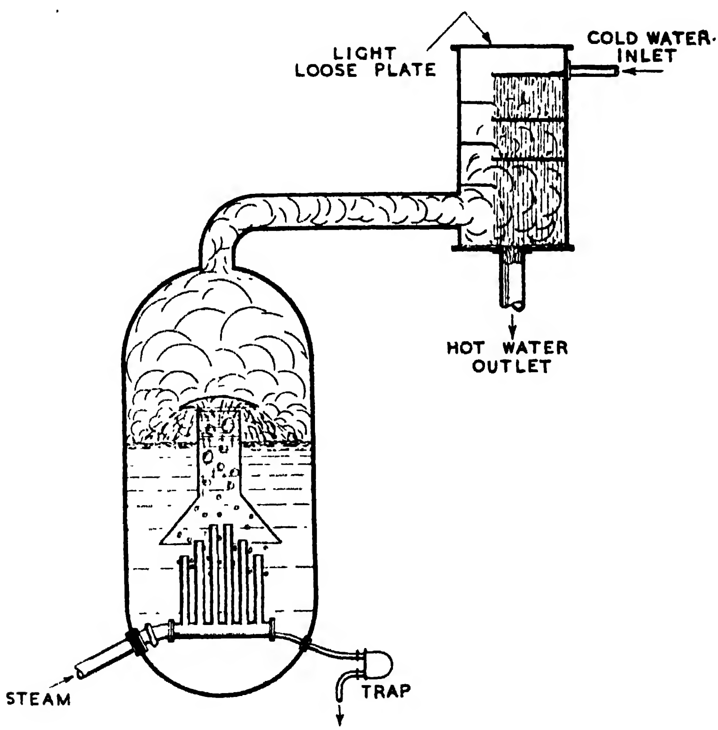


FIG. 234. COLLECTION OF BREWERY COPPER VAPOUR WITH SPRAY CONDENSER

In the author's factory a very long spray condenser collected the vapour heat from about a dozen different sources, all of which were separately ducted into the condenser, which was naturally christened the " Centipede ". Nearly all these vapours have since been promoted to higher grade uses and the poor centipede has suffered successive amputations so that now it is a mere biped, and looks like becoming a monopod.

409. VACUUM FLASHING. In Sections 401 and 402 steam was flashed off a liquid under vacuum. In the application described in those Sections the primary object of the flashing was to cool ; the secondary object was to concentrate ; the flashing was not being done primarily to recover the heat. In Section 402 part of the flash heat was recovered, but only for purposes of making up the temperature lost by excess flashing.

There is no reason why flash steam should not give up its heat to another liquid in a condenser at very low temperature (or high vacuum) provided there is some suitable low temperature liquid that needs heating. The plant described in Section 402 is now being altered so as to recover the heat rejected in the jet condenser, which is being replaced by a surface condenser.

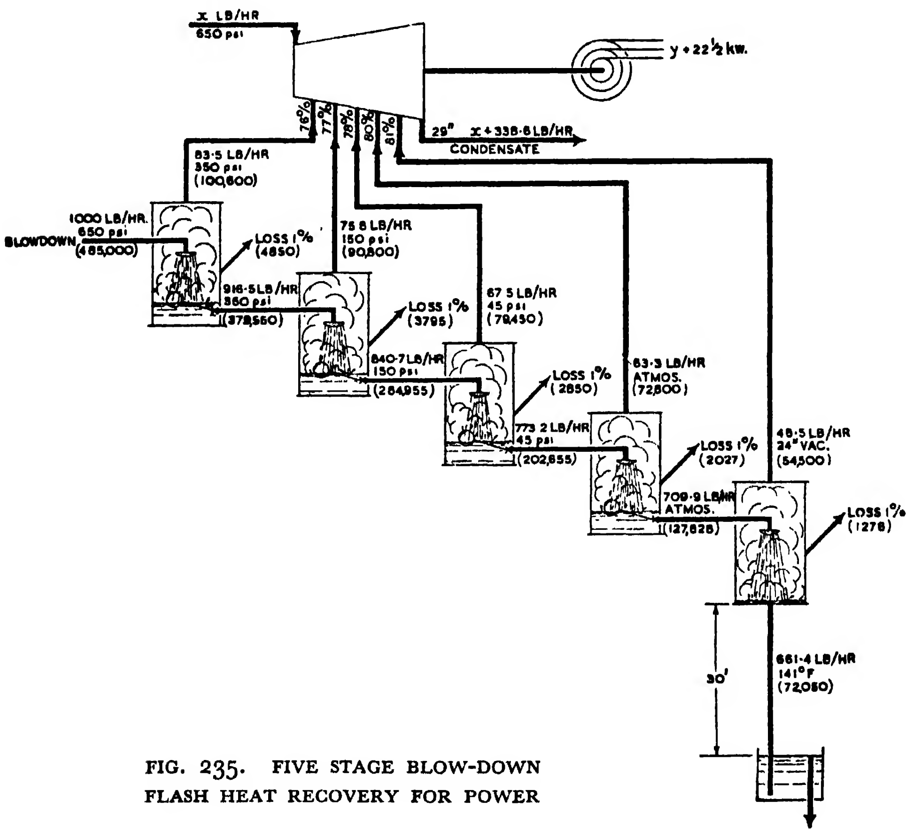


FIG. 235. FIVE STAGE BLOW-DOWN FLASH HEAT RECOVERY FOR POWER

The most economical way of using flash steam at any temperature is to pass it into an existing heating surface, instead of making a special little plant for its use. If there exists a heating surface heated by steam below atmospheric pressure, any water that is being thrown away and that is hotter than the low pressure heating steam, should be connected to the low pressure heating surface and allowed to give up its excess heat as flash steam. This subject is dealt with in more detail in Chapter 17.

410. FLASH FOR POWER. While it is ideal to capture the latent heat in the flash steam, it may not always be possible. There is no sense in installing an elaborate arrangement for collecting flash steam and then finding that there is no way in which the flash heat can be used. But steam is steam whether it be flash or virgin.

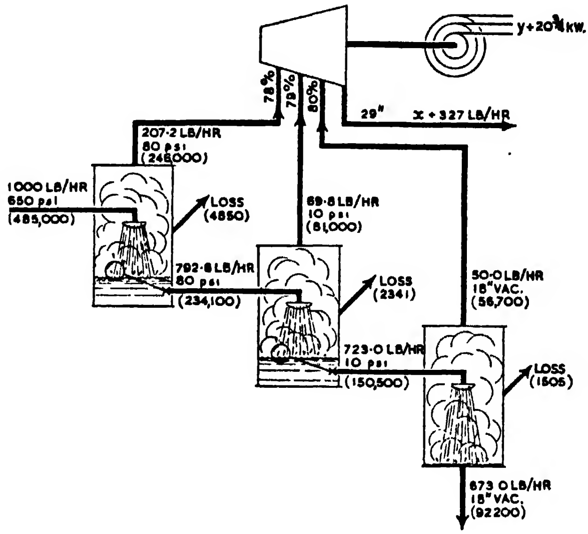


FIG. 236. THREE STAGE BLOW-DOWN FLASH RECOVERY FOR TURBINE BLEED HEATING

Fig. 235 shows a quintuple expansion flash system taking the heat out of high pressure boiler blow-down and passing the steam into a power station turbine. The system is somewhat similar to that shown in Fig. 229 but the flash steam instead of being used for process is used for power production. The turbine has been converted into a multiple mixed pressure turbine with five very small low pressure instalments. A loss of 1 per cent. of the total heat passing through each flash tank has been assumed. Now any steam that passes through a turbine works at the efficiency of all the rest of the steam, so that although these flashes are relatively very small they are used very efficiently and the power obtainable adds up to the truly remarkable total of 22½ kW, while the flash provides a welcome addition of 338½ lb. of distilled water. If the blow down was 1 per cent., this system reduces the actual water loss by blowdown to 0·66 per cent.

411. FLASH FOR BLEED HEATING. The suggestion put forward in the last section demands a multitude of little connections to the turbine. Now a modern turbine has bleed points for regenerative heating. In Section 400 the flash from high pressure blowdown was shown going at two pressures to process. In the modern power station the regenerative feed heating by bled steam is a very definite process.

From the power point of view it does not matter whether the flash steam is used direct into the turbine or direct into the bleed feed water heaters. If it goes into the feed water heaters it will reduce the amount of bled steam and the unbled steam can pass right through the turbine to generate extra power.

In Fig. 24, Section 88, Chapter 2, a regenerative cycle is described in which there are three bleed points at about 80 psi, 10 psi and 18 in. vacuum. Let us see what we can get from flash steams from the blowdown when it is passed through three flash tanks at these pressures. For simplicity it is assumed that the flash steam goes into the turbine. In practice it would go into the feed heaters and would replace an equal amount that would remain in the turbine instead of being bled out. There is no need to work through the arithmetic—the quantities are shown on Fig. 236. We see that by using three stages instead of the five used in Fig. 235, we can get $20\frac{3}{4}$ kW and we secure 327 lb./hr. of condensate. The loss by the use of three stages only and of going down to 18 in. instead of 24 in. is only $1\frac{3}{4}$ kW and $11\frac{1}{2}$ lb./hr. of condensate. Clearly this is the method to use. It is also a good example of the value of investigating a simple plant before embarking on a complicated one. Both these flash-cum-turbine examples may be considered far-fetched. Perhaps they are ; but they are good exercises in flash technique and the scheme in Fig. 236 may well have practical application.

412. JET CONDENSERS. An ordinary jet condenser is as a rule not a device for saving heat but for throwing it away as rapidly and completely as possible. However, jet condensers (and surface condensers) are useful and necessary plants, and this is probably the most appropriate place for their discussion.

Jet condensers are condensers in which the vapour to be condensed and the cooling water come into direct contact without the intervention of a heating surface (or cooling surface). There are a number of different types :—

1. Wet condensers.
2. Dry condensers.
    - (*a*) Parallel current.
    - (*b*) Counter current.
        - (i) Barometric.
        - (ii) Low level.

(Ejector condensers.)

Theoretically any combination of three qualities having either alternative in the first three groups is possible, but in practice there are certain combinations which are never used and would be almost impossible to arrange.

A " wet " condenser is one in which the condensing water and the incondensible gases are removed together by means of a " wet " vacuum pump. The gases are therefore at the hot temperature of the condenser—the water outlet temperature.

A " dry " condenser is one in which the condensing water and the incondensible gases are removed separately. The gases are removed from a point near the cooling water inlet and are therefore at the cool temperature in the condenser—approximately that of the input cooling water.

In a "parallel" current condenser the vapour to be condensed and the cooling water enter at the top and flow together through the condenser.

In a "counter" current condenser the water enters at the top and the vapour enters at the bottom and the two flow in opposite directions.

In a "barometric" condenser the water outlet pipe at the bottom of the condenser is made of such a height that the water column is higher than that corresponding to the vacuum it is proposed to produce in the condenser. The end of the "barometric leg" is sealed in an "atmospheric tank" to prevent the vacuum being broken in the event of the outflowing water being insufficient to keep the barometric leg full of water. (See Section 308 and Fig. 149. See also Fig. 239, Section 417.)

A "low level" condenser is used where there is insufficient height to permit the barometric arrangement, and cooling water has to be removed from the condenser by means of a pump.

The "ejector" condenser is a combination of parallel current condenser and vacuum pump.

413. VACUUM OBTAINABLE IN JET CONDENSERS. The vacuum in a jet condenser is dependent on two things only, the temperature of the cooling water at the maximum temperature that it reaches after absorbing all the heat in the vapour, and on the amount and consequently the partial pressure of any air or incondensible gas in the condenser. The temperature of the hot cooling water depends on its original temperature and the amount that is admitted to the condenser. The amount of air depends on the way in which the air is removed.

The pressure in a condenser is the sum of the partial pressures of the water vapour and the air. If there were no air in the condenser, the pressure—or vacuum—would depend only on the temperature, and the temperature would depend only on the amount of cooling water put in and on its original temperature.

Suppose we wish to condense 1 lb. of air-free vapour at 21 in. vacuum. The temperature corresponding to this vacuum—the steam table tells us the absolute pressure is 4·41 psi.a.—is 157° F. and this is the temperature to which the cooling water is raised. The total heat at 21 in. vacuum is 1,129 Btu/lb. Let us assume that the cooling water is at 60° F. and contains 28 Btu/lb. The output cooling water at 157° F. contains 125 Btu/lb. Then

If x is the weight of the cooling water

$$\begin{aligned}(1 \times 1{,}129) + (x \times 28) &= (1 + x)\,125\\ 1{,}129 + 28x &= 125 + 125x\\ 1{,}004 &= 97x\\ 10{\cdot}35 &= x \text{ lb. water/lb. vapour.}\end{aligned}$$

If condensation is required at atmospheric pressure

$$\begin{aligned}(1 \times 1{,}151) + (x \times 28) &= (1 + x)\,180\\ 1{,}151 + 28x &= 180 + 180x\\ 971 &= 152x\\ 6{\cdot}39 &= x \text{ lb. water/lb. vapour.}\end{aligned}$$

A vacuum of 29·5 in. corresponds to a temperature of 58·8° F., consequently cooling water at 60° F. cannot produce such a vacuum. The water must be cooler than the temperature corresponding to the vacuum needed. Let us see how much water at 60° F. is needed for condensation at 29 in. vacuum.

$$(1 \times 1{,}096) + (x \times 28) = (1 + x)\,47$$
$$1096 + 28x = 47 + 47x$$
$$1{,}049 = 19x$$
$$55\cdot2 = x \text{ lb. water/lb. vapour.}$$

In the foregoing examples it has been assumed that the temperature of the inside of the condenser is the same as that of the outgoing cooling water. This is true of a good counter-current dry condenser, so far as measurements with ordinary industrial thermometers—laboratory checked—are concerned.

414. EFFECT OF AIR. The partial pressure of any air present is added to the vapour pressure of the hot cooling water so that it is necessary to have cooler water in the condenser to obtain the same vacuum as can be obtained with an air-free condenser. We can tabulate for the three pressures dealt with in Section 413 above, assuming 10 per cent. of air as follows :—

Vacuum Required	*Corresponding Temperature*	*psi.a.*	*Partial Air Pressure*	*Partial Vapour Pressure*	*Corresponding Temperature*	*Lb. Water per lb. Vapour*	*Lb. Water per lb. Air-free Vapour*
Atmos.	212	14·69	1·47	13·22	207	6·63	6·39
21″	157	4·41	·44	3·97	153	10·7	10·35
29″	79	·49	·05	·44	76	65·6	55·2

We see from this that at atmospheric pressure or at moderate vacua the presence of a little air does not make very much difference to the amount of cooling water needed ; but that as the vacuum rises and the condenser temperature gets nearer to that of the incoming cooling water the effect of air is serious. Let us see how much air would be required to prevent water at 60° F. producing a vacuum of 29 in.

The vapour pressure of water at 60° F. is ·256 psi.a. The pressure equivalent of 29 in. is ·49 psi.a., so that $\frac{(\cdot49 - \cdot256)\ 100}{\cdot49} = 48$ per cent. of air would make the achievement of 29 in. impossible with water at 60° F.

Table LIX shows the amounts of cooling water at 60° F. needed to produce various vacua with air percentages varying between 0 and 30 per cent. and with temperature differences between condenser temperature and outgoing water temperature of between 0 and 10° F.

415. PARALLEL CURRENT CONDENSERS. In a parallel current condenser the cooling water and the vapour go in together. The hot cooling water containing the condensate comes out together with the air. If there were no air the vacuum in the condenser would correspond exactly to the

vapour pressure appropriate to the temperature of the outflowing water. From Table LIX we see that at atmospheric pressure a condenser with no air will require 6·4 lb. minimum of cooling water per lb. of vapour, while at 28 in. vacuum at least 25·3 lb./lb. will be needed.

TABLE LIX. POUNDS OF JET CONDENSER WATER NEEDED FOR DIFFERENT VACUA AND FOR DIFFERENT AIR PERCENTAGES AND TEMPERATURE DIFFERENCES

LB. WATER AT 60° F. PER LB. VAPOUR

PER CENT. AIR BY VOLUME	0 PER CENT.			10 PER CENT.			20 PER CENT.			30 PER CENT.		
TEMP. DIFF.	0° F.	5° F.	10° F.	0° F.	5° F.	10° F.	0° F.	5° F.	10° F.	0° F.	5° F.	10° F.
VACUUM												
Atmos.	6·4	6·6	6·8	6·6	7·0	7·2	6·9	7·2	7·5	7·3	7·6	7·9
5-in.	6·8	7·1	7·4	7·1	7·4	7·7	7·5	7·8	8·1	8·2	8·6	9·0
10-in.	7·5	7·8	8·1	7·8	8·1	8·5	8·1	8·5	8·9	8·6	9·0	9·5
15-in.	8·3	8·7	9·2	8·7	9·2	9·7	9·1	9·6	10·2	9·7	10·3	10·9
20-in.	9·9	10·5	11·1	10·4	11·0	11·7	11·0	11·6	12·4	11·8	12·5	13·4
21-in.	10·6	11·2	11·9	10·8	11·5	12·3	11·5	12·2	13·1	12·4	13·2	14·2
22-in.	11·0	11·6	12·4	11·5	12·2	13·1	12·2	13·1	14·0	13·0	14·0	15·1
23-in.	11·6	12·4	13·3	12·4	13·2	14·2	13·0	14·0	15·1	14·0	15·1	16·3
24-in.	12·5	13·4	14·3	13·2	14·2	15·3	14·2	15·3	16·6	15·3	16·6	18·1
25-in.	13·8	14·8	16·1	14·6	15·8	17·2	15·7	17·1	18·8	17·1	18·8	20·7
26-in.	15·7	17·1	18·8	16·8	18·4	20·3	18·0	19·9	22·1	19·4	21·6	24·2
27-in.	18·7	20·7	23·1	20·2	22·5	25·4	22·0	24·7	28·2	24·1	27·4	31·7
28-in.	25·3	28·9	33·8	27·3	31·6	37·4	30·6	36·0	43·8	36·0	43·7	55·4
28·5-in.	32·6	38·8	47·8	37·3	45·6	58·6	41·8	52·5	70·3	52·4	70·2	105·8
29-in.	55·2	75·3	117·6	65·7	96·0	176·8	87·8	151·1	531·5	131·9	353·2	—

Now air or other gas is always present in a condenser. Air or gas is dissolved in the liquid from which the vapour is springing. Leaks are present in greater or lesser degree. The cooling water always carries some air in solution. This air must be pumped out of the condenser by means of some kind of vacuum pump. Now if a pump is applied to a vessel containing hot water and no air, the vessel will be exhausted until the pressure inside corresponds to the vapour pressure of the water at its particular temperature. Any attempt to get a greater volume, that is to reduce the pressure, by increasing the capacity of the pump, will fail because the water will just flash and the pump will simply pump vapour, of which there will be an almost unlimited supply.

When air is present, the vacuum in the vessel will correspond to the sum of the partial pressures of the air and the water vapour, and, at the outlet of the condenser, the air represents a much larger proportional volume than that in the input vapour.

There are therefore two limiting factors which control the vacuum in condensers where the air is removed in contact with the hot water. One is the partial pressure of the air ; the other is the fact that the pump will try to re-evaporate some of the steam from the water. For a given water supply and pump capacity there is a certain temperature, which cannot be exceeded, and

which is lower than that corresponding to the vacuum by quite an appreciable amount, due to the air partial pressure and which compels a margin between the vapour pressure of the water and the actual pressure.

416. COUNTER CURRENT JET CONDENSERS. In a counter current jet condenser things are very different. The vapour enters at the bottom ; the water is sprayed in at the top and flows out at the bottom ; the air is removed at the top. This arrangement at once brings two great advantages. The air is properly separated and does not add itself and its partial pressure to the vapour. The temperature of the cooling water, as it leaves the condenser, can therefore be almost the same as that of the incoming vapour. At the point where the air is removed there is air only and cold air at that ; it is of course saturated with moisture. All that is needed is a pump of sufficient capacity to remove the cold air. It is therefore always possible to get a given vacuum from a counter current condenser with less water and a smaller vacuum pump than from a parallel current condenser. No parallel current condenser can operate at, or near, the vacuum corresponding to the water temperature. A counter current condenser can and does operate in such a way that ordinary industrial thermometers register no difference between the temperatures of the input vapour and output water.

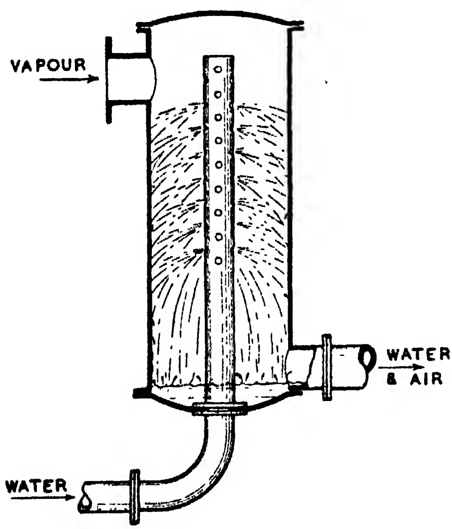


FIG. 237. PARALLEL CURRENT JET CONDENSER

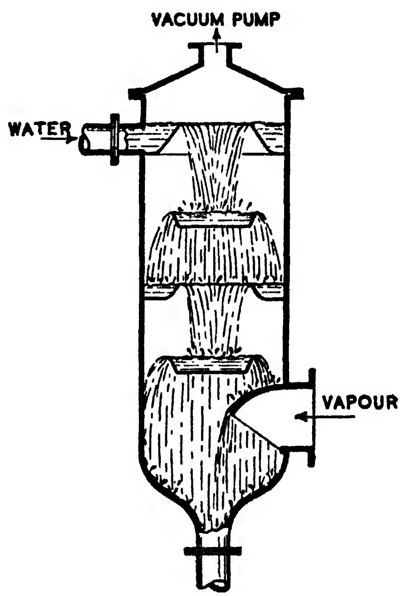


FIG. 238. WATER CURTAIN COUNTER CURRENT JET CONDENSER

417. TYPES OF JET CONDENSER. The parallel current condenser is not important. It should be avoided if possible. It is extravagant in water and in vacuum pumping power. One old design is shown in Fig. 237.

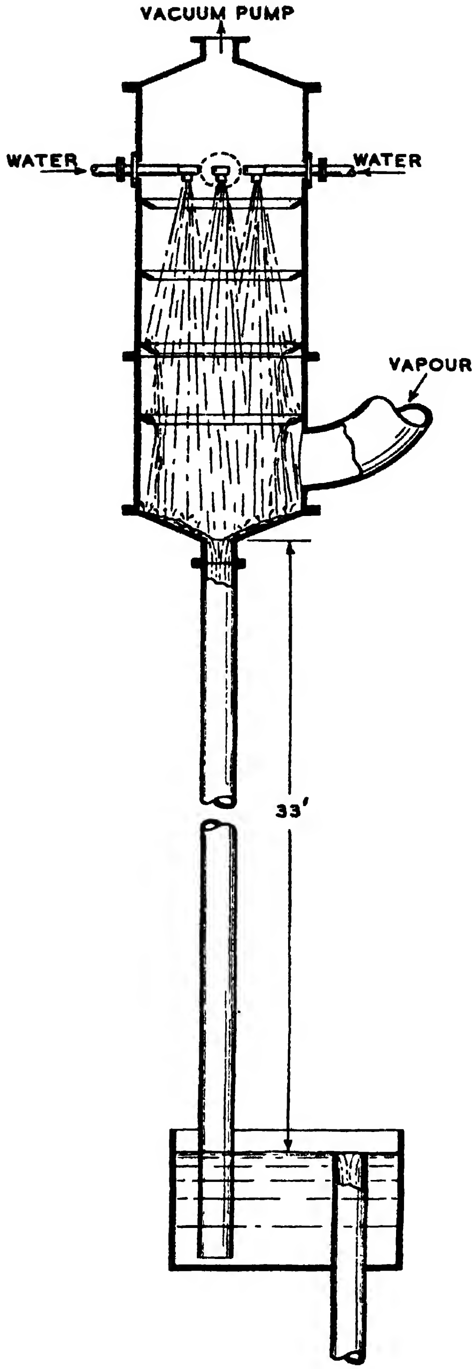


FIG. 239. WATER SPRAY COUNTER CURRENT JET CONDENSER

Counter current jet condensers are of three broad types; the water curtain; the spray nozzle; and the tray. A type of water curtain condenser is shown in Fig. 238. Many of the proprietary makes of condenser are of this type. The water overflows a weir and falls as a tubular curtain on to one or more baffles or saucers where it is redirected as another tubular curtain. The curtain is seldom really continuous, but even so, there is a definite resistance to the vapour flow as the vapour must break the curtain to get through.

The spray nozzles in Fig. 239 break the water up into drops and the water control brings successive nozzles into or out of operation so as to maintain proper drop formation at all loads.

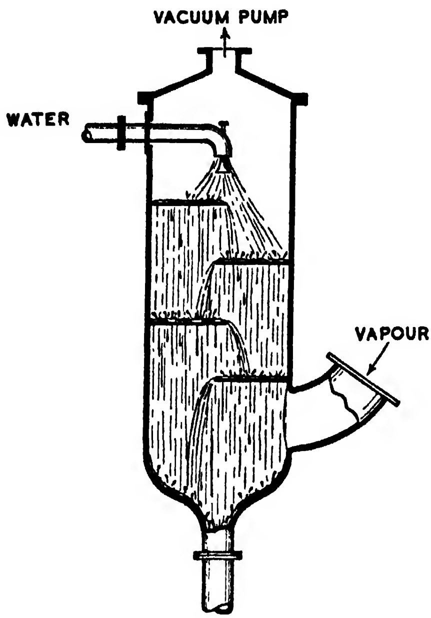


FIG. 240. PERFORATED TRAY COUNTERCURRENT JET CONDENSER

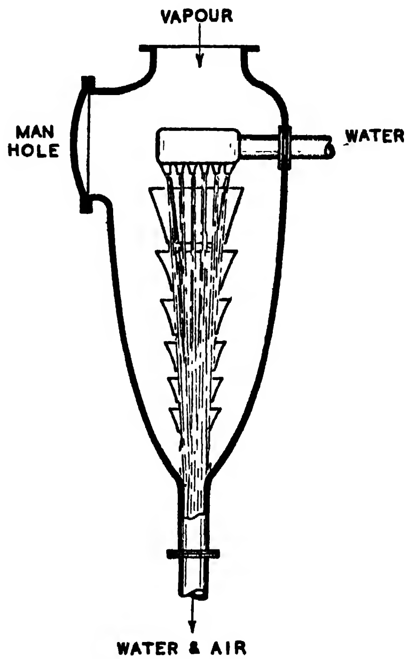


FIG. 241. EJECTOR CONDENSER

The tray condenser, shown in Fig. 240, is crude but is, in the author's experience, quite the best form of jet condenser. The trays are made of perforated cast iron and are shelves which are just over half a diameter in area. The vapour has a free unrestricted path. It has not got to force its way through tough curtains of water. The water is delayed in its passage through the condenser. It so has plenty of time to take up as much heat as it can, unlike the water in the condenser shown in Fig. 239, where the water can drop without hindrance straight out of the condenser. The tray condenser is the only condenser in which it has been found possible (in the author's works at any rate) to have a water outlet temperature equal to the vapour inlet temperature.

418. EJECTOR CONDENSERS. These are hybrid machines. They combine vacuum pump with parallel current jet condenser. They must use more water than a plain parallel current condenser, because the water acts as vacuum pump as well as coolant. Where water is very abundant and cheap they may fill the bill.

Fig. 241 shows the principle on which they work. The vapour enters at the top, and water is sprayed through a ring of special nozzles or jets into a throat which is built up of a series of ejector cones. The water drags the air out by suction through the ejectors. These condensers are not economical on varying loads as a reduction in water supply to the nozzles reduces the ejector efficiency.

All three forms of counter current barometric condenser act as ejector condensers to a certain extent. Tray condensers in the author's factory give a slightly better vacuum (when there is no vapour entering the condenser) with the cooling water flowing through the condenser and the vacuum pump working, than with the vacuum pump connected to a dry, but water sealed, condenser. This means that not only does the water eject all the air brought in with itself but does a little extra vacuum pumping as well.

419. LOW LEVEL CONDENSERS. These should never be used if there is head room sufficient for a barometric condenser. All the water must be pumped out. This, from a power point of view, should not matter, as all the water in a barometric condenser must be pumped 30 feet higher. But the pump that extracts the water in a low level condenser is generally a reciprocator and doubles the part with the vacuum pump. It is possible for an ejector condenser to be a low level condenser.

Every low level condenser must be provided with a float operated vacuum breaker. If for any reason the condenser starts filling up with water the vacuum must be broken to prevent the condenser water flowing back into the plant. To quote the inimitable Mr. McIntosh : " If, from any combination of circumstances, the condenser be overloaded with water, the latter, instead of being removed altogether by the pump, flows back into the vacuum pan, and, mingling with the sugar solution, proves very detrimental to the ultimate result."

* * *

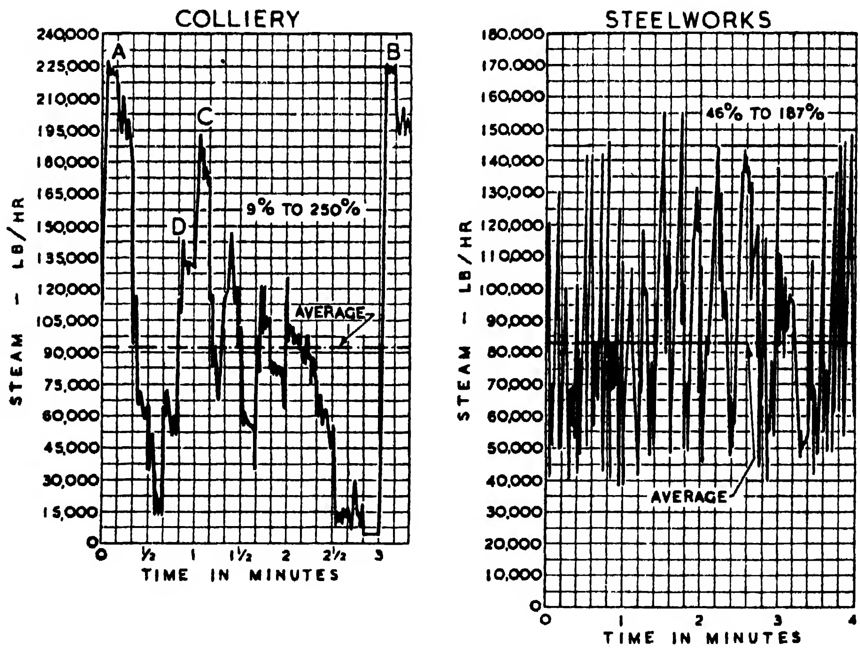
CHAPTER 15

PEAK LOADS

Their loftiest peaks most wrapped in clouds.
BYRON. *Childe Harold.* 1816.

PEAK loads and their inevitable companions, valleys, are amongst the greatest causes of waste, both of fuel and of time, in many types of factory. Production can be increased, fuel consumption reduced and often quality improved by cutting off the peaks and filling in the valleys. A little effort applied to smoothing the peaks will often make plant serve well when it was thought that extra plant must be bought.

420. SOME TYPICAL INDUSTRIAL LOADS There is hardly a factory using steam where the demand is steady. In many industries the load fluctuates wildly. Sometimes the peaks are measured in seconds, sometimes in minutes and sometimes in hours. Figs 242 and 243 show the steam loads of four different plants as given by E. G. Ritchie. These are not exaggerated, they can be confirmed by anyone who has access to such industries.



a b

FIG. 242. TYPICAL SHORT TERM PEAKS

421. SHORT TERM PEAKS. Fig. 242a shows a colliery steam load measured over a period of just over three minutes. The steam flow varies from 5,000 to 225,000 lb./hr. The two very large peaks at A and B are due to three winders starting simultaneously. Now it is obvious that it is quite impossible for any

boiler plant to meet such a steam demand properly. The firing rate must be kept steady at average load and when these wild peaks come along they must be met by a drop in pressure, which will cause the boiler water to flash and thus help to meet the peak. As such sudden steam draws cause a grave risk of heavy carry-over, it will be necessary to keep at least 50 per cent. more boilers in operation than would have been needed to carry the average load had the demand curve been smooth.

Fig. 242b shows the steam load in a steel works. Here the peaks are caused by ingots, blooms and billets passing through the rolls, or by steam hammer blows or press squeezes. Again the peaks are very short and quick so that it is quite impossible for any boiler plant to follow them, they can only be met by allowing momentary pressure drops on the boiler range.

422. **LONG-TERM PEAKS.** Fig. 243a shows the steam demand in a small dye works. Here the peaks and valleys are of quarter to half an hour in length. Peaks and valleys of this length must be met by the boilers, because the length and severity of the pressure fluctuations would be intolerable. The peaks are caused by the intermittent putting on and off the dye becks or vats.

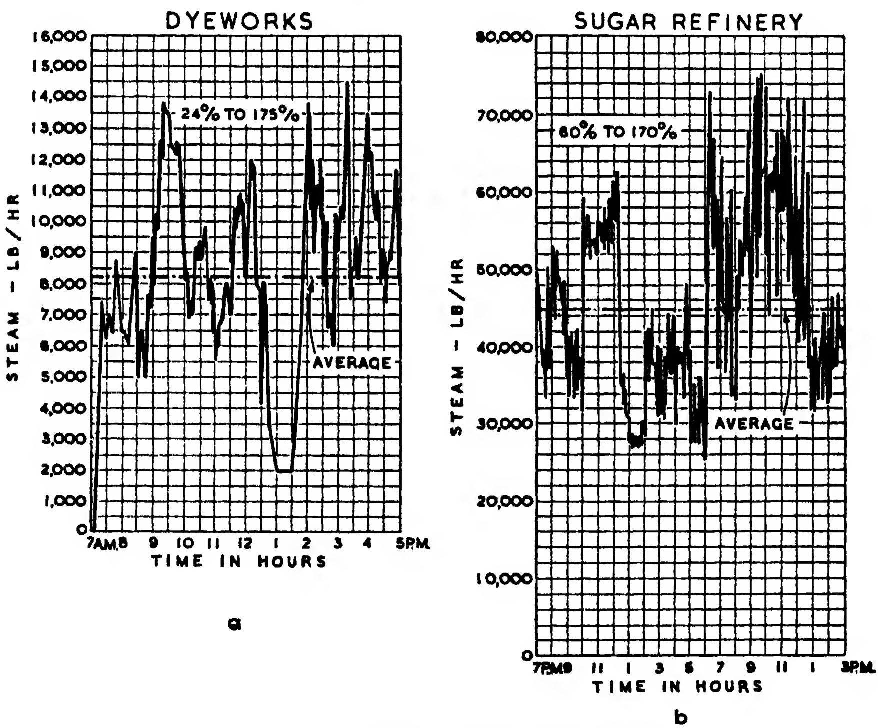


FIG. 243. TYPICAL LONG TERM PEAKS

Fig. 243b shows the steam demand in a small sugar refinery. The main peaks are caused by the intermittent operation of the vacuum pans which work on cycles varying from 2 to 12 hours. Superimposed on these major peaks are

small peaks from blowers and heating coils. These small peaks are too short and rapid to be met by the boiler, but the big long peaks must be handled by the boiler plant.

423. **EFFECT OF PEAKS ON BOILER OUTPUT.** It is all very well to say that such and such peaks must be met by the boilers. What happens when the boilers try to meet peaks of the kind we have seen in Fig. 243? Ritchie gives the following excellent description of what happens in an ordinary small boiler house.

"Fig. 244 is a diagram showing the rate of steam demand (not steam flow) of a factory as the full line, the rate of steam production by the broken line and the steam pressure by line C.

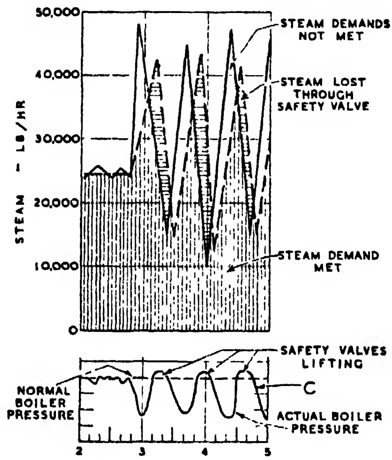


FIG. 244. PEAKS THAT THE BOILER PLANT MISSES AND CONSEQUENTIAL BLOW-OFFS

Between 2·0 and 2·45 it is assumed that the conditions are steady and that a rate of firing has been established corresponding to the steam demand, so that the pressure is reasonably well maintained. At 2·50 a peak develops causing the boiler pressure to drop. With this indication that he is short of steam, the fireman increases his rate of firing and opens his dampers. The boiler is, however, slow to respond and the boiler pressure continues to fall. Meantime the peak demand has reached its maximum at about 2·55 and has begun to fall off. At 3·05 the safety valves lift, and, with this indication that he is making more steam than is required, the fireman reduces his rate of firing and closes his dampers. This does not take effect immediately, and, in any case, the steam demand is falling. Consequently the safety valves remain open until about 3·20 when the rate of steam production is again equal to the demand. At this point, however, another peak demand starts and the chase begins again—and so it may go on all day.

The white areas in Fig. 244 represent steam demands that have not been met—they would be missing altogether on a steam meter chart, except for the

amount of flash steam produced from the boiler water by the drop in pressure due to the peak demand. The horizontally hatched areas show the amount of steam blown off by the safety valves. The vertically hatched area shows the steam demand which is actually met."

424. EFFECT OF PEAKS ON BOILER EFFICIENCY. A boiler requires constant care and attention in order that combustion may be efficient. Unless load conditions are steady it is impossible to adjust coal supply, fire thickness, draught, air supply, etc., so as to attain efficiency. If the fireman is waging continual war on fluctuating steam pressure, chasing a peak demand with falling pressure one minute, damping down a rising pressure the next, it is quite impossible for him to pay any attention to combustion. All he can do is to get maximum heat liberation now and damp everything down then. The probable boiler loss from such a state of affairs may easily be as much as 8 per cent. to 12 per cent., however good the condition of the brickwork, dampers, etc. On the top of these losses is the loss from the safety valves. This may be much greater than is often appreciated, anything from 1 per cent. to 10 per cent. It is pretty certain that a hand-fired boiler house, where meeting the peaks and valleys occupies the major part of the fireman's attention, will be between 10 per cent. and 15 per cent. less efficient than a similar plant on steady load where the fireman can devote all his time and effort to getting efficient combustion.

425. CUMULATIVE EFFECT OF PEAKS. When one or two pieces of plant start up together the steam pressure in the pipe feeding these plants drops. The operators in the neighbourhood at once open their valves to maintain the rate of heating on their plants with the reduced pressure. This pulls down the pressure still more and the original offenders in their turn open up still more and a proper steam chase starts. The lower the pressure the less steam can be carried by a given steam pipe, so that to get the necessary flow a bigger pressure drop will be needed. This causes a still further lowering of the pressure.

Once a small peak starts there are several influences at work tending to aggravate it. Similarly, when the peak starts to become a valley everyone starts closing their valves against rising pressure, pipes carry more steam with less pressure drop, so that the valley tends to become deeper.

426. EFFECT OF PEAKS ON STEAM FLOW. In Section 174, Chapter 5, we found that to carry 6,000 lb. of saturated steam an hour at 60 psi.g. along a pipe 450 feet long (effective length 531 ft.) we should need a pipe 4 in. in diameter which would cause a pressure drop of 1·5 psi per effective 100 ft. or a total drop of 8 psi. This would mean that the pressure at the plant would be 52 psi.

What happens when, due to a peak demand in another department, the steam pressure in the main drops from 60 to 50 psi ? We must guess the mean volume of the steam as 6·25 cu. ft./lb. Fig. 49 shows that the pressure drop per 100 ft. will be 1·6 psi. So that the pressure drop along the whole pipe will be 8·5 psi instead of 8 psi, and the lower the pressure the greater must be the pressure drop.

427. EFFECT OF LOWERED PRESSURE ON PROCESS. Suppose a vat contains 500 gallons of water to be heated from 60° F. to 210° F. and then held at that temperature for 2 hours. We will assume that there is 2 ft. 8 in. of liquid in the vat and that steam at 4 psi.g. is fed to the blowers. (Steam at 4 psi is lower than the pressures used in most dye works. But 4 psi should be sufficient and there will be much less waste than with higher pressure steam.) The vat is 4 ft. × 4 ft. × 9 ft. The sides and bottom have 1 in. of lagging but the top is open.

The heat required for heating up the water will be

$$210 - 60 = 150 \text{ Btu/lb.}$$

Steam at 4 psi contains 1,155 Btu/lb.

The steam required will be found thus :—

$$\begin{aligned}(x \times 1{,}155) + 5{,}000\,(60 - 32) &= (5{,}000 + x)\,(210 - 32)\\ 1155x + 140{,}000 &= 178x + 890{,}000\\ 977x &= 750{,}000\\ x &= 768 \text{ lb.}\end{aligned}$$

The heat loss at 210° F. can be found from Table XXII as :—

lagged .. .. 50 Btu/sq. ft./hr.
bare .. .. 325 Btu/sq. ft./hr.

The lagged surface is $(4 \times 4 \times 2) + (4 \times 9 \times 3) = 140$ sq. ft.

Unlagged surface is $4 \times 9 = 36$ sq. ft.

The open top will permit evaporation, so that it is quite certain that the heat loss will be much more than the 325 given by Table XXII. It will probably be well over 1,000 Btu/sq. ft./hr. (see Section 464).

The total heat loss will be

$$(140 \times 50) + (36 \times 1{,}000) = 43{,}000 \text{ Btu/hr.}$$

The heat needed to make up for the losses will be provided by the latent heat in the 4 psi steam plus the difference in sensible heat in the steam and the tank contents, or, at these low temperatures, the difference between the steam temperature and the vat temperature.

Latent heat of 4 psi steam is 963.

Sensible heat difference is $225 - 210 = 15$.

The steam required to make good the losses will be

$$\frac{43{,}000}{978} = 44 \text{ lb. steam/hr.}$$

In practice the steam will probably be wet and there will be draughts so that a safe figure to take would probably be 60 lb./hr. This may even be an underestimate.

If the steam pipes are such as to pass 500 lb./hr., the actual time needed for the process cycle can be estimated with the help of Figs. 245 to 247.

The steam needed to make good the losses is 60 lb./hr.

The maximum steam supply available is 500 lb./hr.

When the vessel is cold, as at A in Fig. 245, there are no losses.

As the vessel is warmed the losses increase until they reach the equivalent of 60 lb./hr. at B when the vessel is up to temperature.

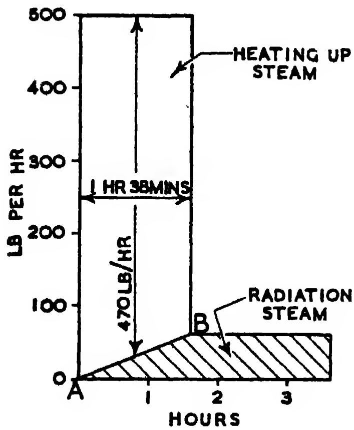


FIG. 245. NORMAL BATCH DYEING OPERATION

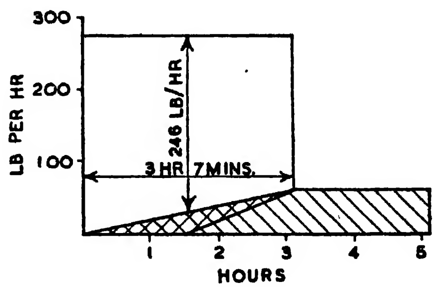


FIG. 246. DYEING OPERATION WITH REDUCED STEAM PRESSURE

If the liquid is capable of condensing the steam at the full rate of possible supply the cycle is fairly shown in Fig. 245.

The actual rate of steam supply available for heating up the liquid from cold is 500 lb./hr. less half the loss-balancing steam or 500 − 30 = 470 lb./hr.

The amount of steam needed for heating up has already been found to be 768 lb.

So that the time taken for heating up will be

$$\frac{768}{470} = 1 \text{ hr. } 38 \text{ min.}$$

The total processing time will be 3 hr. 38 min. assuming that the actual dyeing operation takes 2 hours.

Now the pressure drop limiting the steam flow in the blower is 4 psi less the hydrostatic head of 1 psi, or 3 psi.

If the pressure in the steam pipe drops to 2 psi due to a peak demand elsewhere there will only be a pressure drop of 1 psi available in the blower.

The mean volume of the steam at 4 psi is

$$\frac{21\cdot4 + 25\cdot2}{2} = 23\cdot3 \text{ cu. ft./lb.}$$

The mean volume of the steam at 2 psi is

$$\frac{23\cdot8 + 25\cdot2}{2} = 24\cdot5 \text{ cu. ft./lb.}$$

We can estimate the reduction in steam flow from Fig. 49, Section 174, thus :—

500 lb./hr. is 8·3 lb./min.

On Fig. 49 find the intersection of 8·3 on the horizontal scale with 23·3 cu. ft./lb. on the vertical scale.

Run down the reference line till the 3 psi pressure drop on the vertical scale is reached.

This gives us an equivalent pipe diameter of $1\frac{3}{4}$ in.

Now find the intersection of this pipe diameter with 1 psi pressure drop.

Run up the reference line to the intersection with 24·5 cu. ft./lb.

This intersects a flow on the horizontal scale of 4·6 lb./min. or 276 lb./hour.

The heating up time will be $\dfrac{768}{(276 - 30)} = 3$ hr. 7 min.

The total cycle time will be 5 hr. 7 min. and is shown in Fig. 246.

Output will be reduced by 40 per cent.

The increased radiation loss is shown by the cross-hatched area in Fig. 246 and amounts to 45 lb. of steam per batch.

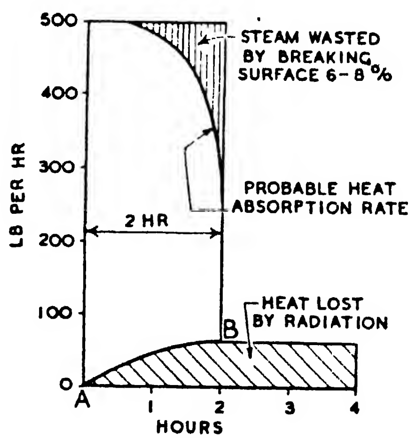


FIG. 247. DYEING OPERATION WITH STEAM WASTED BY WIREDRAWN INJECTION

In actual practice probably the conditions will never be quite as shown in Figs. 245 and 246. The heat absorption will probably fall off as the vat heats up. As it is unlikely that there is really proper control much steam will probably break the surface and be wasted. Fig. 247 shows the more probable state of affairs. It shows a possible waste of steam breaking surface of $6\frac{1}{2}$ per cent. It also shows the more likely amount of radiation loss. As the temperature of the vat rises the radiation loss will probably increase by rather more than direct proportion to the temperature drop.

428. THE BAD EFFECT OF PEAKS ON QUALITY. In those processes where exact temperature is essential for proper processing, for example the curing of rubber and many chemical processes, peaks and subsequent valleys cause alternate high and low steam pressures. It will therefore be necessary, in order to maintain the constant required temperature, to work the steam main at a much higher pressure than would have been necessary with a steady pressure, in order to have a good margin. It will also be necessary to have very accurate reducing valves or very conscientious operation.

In many operations too high a temperature is just as detrimental as too low a temperature. Peaks and valleys will either spoil material or will make good processing more difficult, or both.

429. EFFECT OF PEAKS ON ENGINES. The indicator diagram in Fig. 248 shows an engine working with steam at 155 psi.g., line A. If the engine is cut-off governed the power will remain constant when the pressure falls to 125 psi as shown by the chain dotted line B and cut-off will take place at 35 per cent. instead of 25 per cent. This will increase the steam consumption by about 16 per cent. (The steam consumption is not proportional to the cut-off because much of the steam is used to compensate for initial condensation and for filling the clearance space). If the engine is throttle governed and was on full load at 155 psi the power output will drop by 23 per cent. if the pressure drops to 125 psi. This is shown by the broken line C in Fig. 248.

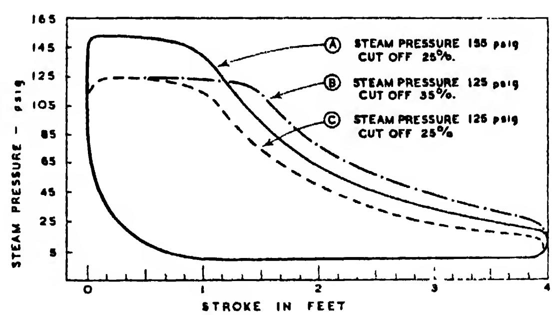


FIG. 248. EFFECT OF REDUCED STEAM PRESSURE ON ENGINE PERFORMANCE

430. THE BAD EFFECTS OF PEAKS. We have seen that peaks and valleys greatly reduce the efficiency of the boiler house ; that they demand a larger boiler house ; that drops in steam pressure reduce output ; that lowered pressures increase radiation losses ; that spoilt material may result ; that correct processing is more difficult ; that engines give less power. It behoves us therefore to do all that we can to smooth off the peaks and fill up the valleys.

431. THE CAUSES OF PEAKS. If we look carefully at Figs. 242 and 243 we can see certain things. In Fig. 242a the two big peaks at A and B were caused, we know, by three winders starting together. By the shape of these peaks we can say with fair certainty that peak C is one winder and peak D is another.

In the steelworks in Fig. 242b many of the peaks are momentary, so short in fact that little or nothing can be done about them, except to have a boiler plant that has generous water capacity and therefore ample steam storage and is sufficiently lightly loaded not to prime at the peak demands.

The dye-works in Fig. 243a shows us little except that the demand is haphazard as might be expected from the nature of the process, but the peaks are of fair duration, they are not momentary like those of the steelworks.

The sugar refinery in Fig. 243b shows long-term peaks such as might be expected from the nature of the principal steam users. There are, however, small short peaks superimposed on the big peaks, and it is very interesting to note that these short peaks are twice or thrice as great on the day shift as at night. This gives a line of investigation at once. These sharp daytime peaks should be easily found and it might well prove possible to smooth them out.

Apart from these obvious things an examination of Figs. 242 and 243 leads to the belief that the peaks are unpredictable and cannot be accounted for by any simple analysis.

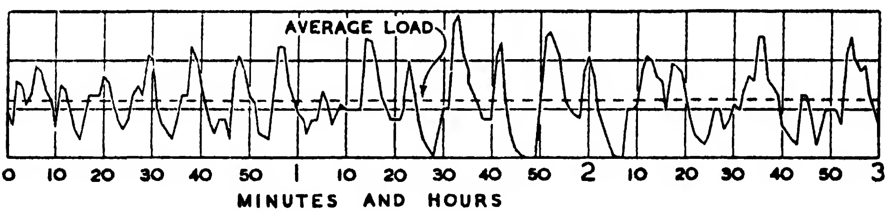


FIG. 249. PEAKY STEAM LOAD FROM INTERMITTENT STEAM USERS

Look at Fig. 249. It looks pretty erratic but close examination shows certain definite characteristics. For the first half hour the peaks are quite small and seem to have a 4 or 5 minutes frequency. Then for half an hour the peaks occur more severely at 9 minutes intervals. The 9 minutes frequency continues for the next hour although the shape of the peaks changes entirely. In the last hour the double peaks of the previous hour have become single peaks at 18 minutes frequency.

Now look at Fig. 250 which shows how the curve in Fig. 249 was derived. The erratic peaky curve of total consumption at A is built up from the perfectly regular operation of the three plants B, C and D. Now if curve A was a true factory steam chart whose erratic behaviour was being investigated due to trouble in the boiler house, an examination of the three individual charts B, C and D would show that they were being run with great regularity, and it might be thought that the plant was operating as well as possible. Let us look more closely.

Plant B heats a batch of material from cold and then cooks it gently for 10 minutes. This plant works on a steady 21-minute cycle. In plant C material is heated up, boiled for 5 minutes, a small addition of cold material is then made. This is then again brought to the boil and the plant shut down. The cycle is

regular at 19 minutes. Plant D is a thermostatically operated drier working intermittently but quite regularly on an "on-off" control on a 9-minute cycle.

If the 19 and 21 minute plants could be brought to a regular 20 minutes, and if the 9 minute plant could work at 10 minute intervals, it might be possible to cut out all the violent fluctuations.

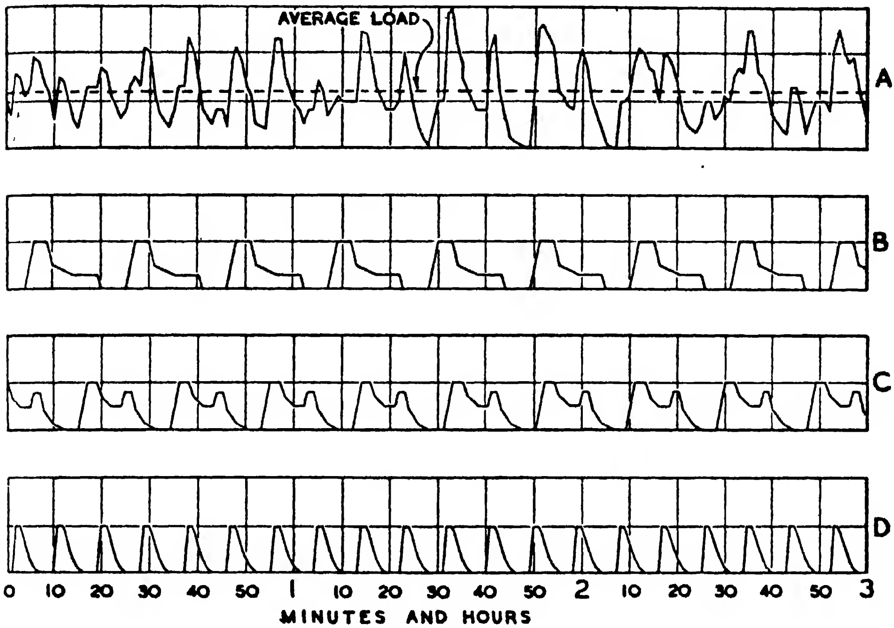


FIG. 250. ANALYSIS OF PEAKY LOAD

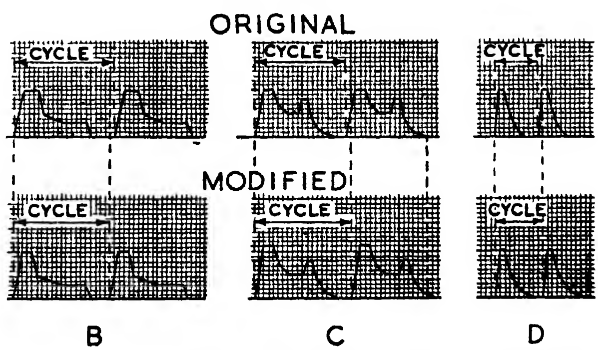


FIG. 251. ADJUSTMENT OF TIME CYCLES ON INDIVIDUAL STEAM USERS

Fig. 251 shows the alteration that will be necessary. If plant B is to work a 20-minute cycle instead of 21 minutes, it will be necessary to cut the size of each batch down by 1/21. Consequently rather less steam will be taken per batch. It will be necessary to cut the idle time from 5 minutes to 4 minutes, and we will assume that this is possible. The change is shown at Fig. 251B.

To run process C on a 20-minute cycle instead of 19 minutes means increasing the size of the batch by 1/19 and consequently the batch steam consumption. We will assume that this is possible. The change is shown at Fig. 251C.

To bring the thermostat D into phase with the others on a 10 minute cycle means decreasing the response of the thermostat. This should prove only too easy. Each operation of the thermostat will call for 1/9 extra steam. The change is shown in Fig. 251D.

Fig. 252 shows these three modified cycles arranged in such a way (a little trial and error will show the best arrangement) that the valleys made by staggering B and C are more or less filled up by D.

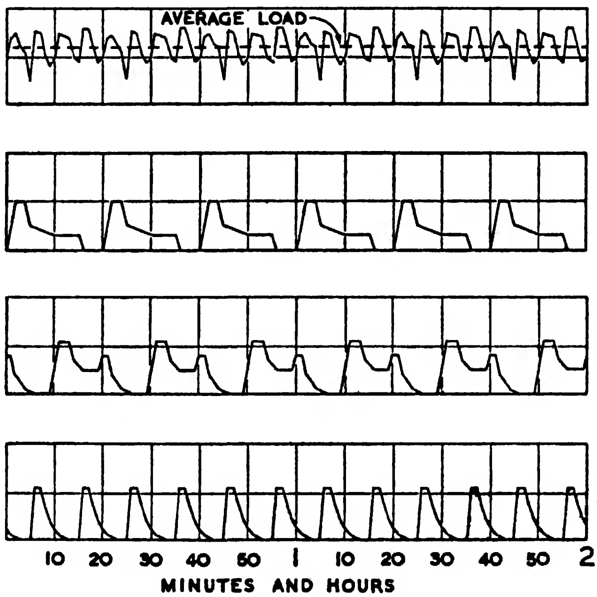


FIG. 252. SYNTHESIS OF ADJUSTED CYCLES REDUCES AGGREGATE PEAKS

432. REDUCING LONG-TERM PEAKS. The remarkable change from A in Fig. 250 to A in Fig. 252 gives us the key to the first and simplest way to smoothing the peaks and the valleys. It is just simple management. A stop-watch, a steam meter if possible, or a good pressure gauge if no steam meter is available, plenty of perseverance and imagination may easily lead to the smoothing out of many of the peaks. It may be argued that Figs. 249 to 252 are not fair. That may well be. They are not meant to be fair. They are meant to show that very difficult peaky consumers can, if other conditions allow, be organised to give a reasonably smooth demand curve.

It is often argued that with plenty of steam consumers their random incidence will iron out any big peaks. This is only true if the number is enormous, if the transmission losses are high and if there is a good base load. Sooner or later all the plants will stop or start together. The greater the number of consumers the greater is the possible peak, but the less often is it likely to occur.

There are few factories that could operate on the tight programme shown in Fig. 252. There are several other ways of improving matters. If peaks are quite inevitable some simple signal system between the plant using steam and the boiler house by means of lights or an engine room telegraph (a very cheap, simple, reliable device which speaks many messages and which deserves much greater use than it gets) will do much to help. Where it is impossible to programme all the plant it will probably be possible to programme some. It is almost always possible to take some plant right out of the regular run of the factory and use it as a peak and valley leveller. Such a plant should if possible be put in charge of the boiler house and might well take the form of the provision of hot water. If the hot water tank is large, the heating of water can be stopped during peak demands and started again when the demand falls off. Much can often be done by steam accumulation or storage and these are dealt with in the next Chapter.

433. REDUCING SHORT-TERM PEAKS. Peaks that are measured in seconds present difficulties that are not associated with long-term peaks. By programming, organisation and signalling, it is possible to level peaks to a great extent if they are measured in minutes or hours. This is almost impossible with short term peaks measured in seconds. A system of signalling between the three winding engines in Fig. 242a might cut the big peaks, but at first sight it would also cut the number of winds. This might, however, not be so. The improvement in boiler performance might lead to such a steadying of the pressure that the number of winds might be increased due to faster winding. Again signals between the rolling mills might be possible, but output might suffer, and while steam heat might be saved, steel heat might be lost. On the other hand good steady steam pressure will reduce rolling time and save steel heat, with less risk of the engine stalling in the middle of a run.

Peaks are sometimes produced by automatic controls of bad design, or even by hand controls of bad design. If the control valve is too big (see Section 205) no automatic or manual control can work other than as an on-off control. This may result in a succession of peak demands on the boiler plant.

While we should do all we can to cut out peaks this should not be done by putting brakes on production or by irksome restrictions. It must never be forgotten that the steam supply is a servant and must not be allowed to become the master. In order that the boiler house may remain a good servant its task should be made as easy as possible. We are probably compelled to accept and deal with the short-term peaks as best we can, but the long-term peaks can often be smoothed away. The cure may be technical or it may be purely managerial. The smoothing of peaks usually saves much fuel and generally improves output and quality of product.

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CHAPTER 16

HEAT STORAGE

> **Laying up in store for themselves a good foundation against the time to come.**
>
> **I. TIMOTHY. VI. 19. A.D. 56.**

IN the last Chapter we have seen how peaks and valleys are bound to lower boiler efficiency and must therefore waste coal ; how they reduce the effective capacity of the boiler house ; how they may spoil the output product ; how they must make the process more difficult ; how they must slow down the process and therefore cause loss of output and higher costs ; how they cause direct waste of steam through safety valves.

Peaks and valleys do not always occur on the high pressure line alone ; they often occur more severely on the low pressure main. Even if peaks can be cut out or diminished on the high pressure line they may be grievous on the low pressure main. Heat storage is often a cure ; it is always a palliative.

434. TYPES OF HEAT STORAGE. The well known types of heat storage are the regenerative and recuperative arrangements on furnaces. We are not concerned with these here. We are only concerned with the storage of steam heat. The principal method of storing steam heat is by transferring its heat to water. This can be done at variable temperature and constant volume, or at constant temperature and variable volume, or at variable temperature and volume. There is another method of storing steam heat and that is to store the actual steam. This calls for more bulky and expensive plant, entails bigger losses of heat, but has other definite advantages. This method has hitherto received comparatively little attention and may deserve much more.

435. STORAGE CAPACITY OF THE BOILER. If we take a small Lancashire boiler, the water content is probably about 600 cu. ft. and the steam content is about 400 cu. ft. Let us consider such a boiler working at 150 psi and let us see what happens when the pressure is allowed to drop to 140 psi and to rise to 160 psi.

Pressure psi.	*Sensible Heat Btu/lb.*	*Latent Heat Btu/lb.*	*Water Volume cu. ft./lb.*	*Steam Volume cu. ft./lb.*
140	333·2	862·5	·0181	2·93
150	338·6	858·0	·0182	2·76
160	343·6	853·9	·0182	2·61

If the water occupies 600 cu. ft. it will weigh 32,965 lb. at 150 psi.

A reduction in pressure of 10 psi will reduce the sensible heat by

$$338 \cdot 6 - 333 \cdot 2 = 5 \cdot 4 \text{ Btu/lb.}$$

This will cause a flash of $\dfrac{5 \cdot 4 \times 32{,}965}{862 \cdot 5} = 206$ lb. of steam.

We have assumed that the steam space has a volume of 400 cu. ft.

The weight of the steam in the steam space at 150 psi is	145 lb.
The weight of the steam in the steam space at 140 psi is	136·5 lb.
Steam available due to expansion	8·5 lb.

So we see that by allowing the pressure to drop by 10 psi we get 214·5 lb. of extra steam from a boiler that is probably producing 7,000 lb./hour, or 117 lb./min. This extra 214·5 lb. represents almost 2 minutes' steaming at normal full load.

Working out the figures for a 10 psi rise in pressure we find that the boiler will absorb 201 lb. of steam, or one and three-quarter minutes' steaming.

A pressure variation of $\pm$ 10 psi, if such a variation can be tolerated, will allow a Lancashire boiler to carry a 200 per cent. load for 2 minutes to meet a peak demand, or zero output for nearly 2 minutes to meet a valley. This shows us that shell boilers are ideally fitted to look after short sharp peaks provided some pressure drop can be allowed. If a pressure drop of 10 psi is more than can be permitted, we can get the same accommodation if we have more boilers working, but the boilers will of course be more lightly loaded.

From the example just given we see that the steam volume has a very poor storage value compared with the water volume. In dropping from 150 psi to 140 psi the water provides 206 lb. of steam whereas the steam space only provides 8·5 lb. of steam. We get ·343 lb./cu. ft. from the water space, but we only get ·021 lb./cu. ft. from the steam space. Volume for volume water provides 16 times the steam storage as does steam space at the particular pressure considered.

While the large water capacity of the Lancashire boiler is a great help in meeting short sharp peaks (although this may be accompanied by priming or carry-over), the Lancashire boiler is not quickly responsive to change of load. For meeting long-term peaks the water-tube boiler is much more suitable and should preferably be used on a peaky load where the peaks and valleys are long.

436. MANY PART-LOADED BOILERS v. FEW FULLY LOADED. From the point of view of storage, to meet fluctuating loads, the bigger the boiler capacity the better. From the combustion point of view, to meet fluctuating loads, the more lightly loaded the boilers, that is to say the more boilers there are on the range, the better, because a given load increase represents a smaller percentage capacity increase on many boilers than when shared by few. It would therefore seem obvious that it is much better to use a number of part-loaded boilers than a few fully loaded, whenever the load contains many short peaks. There are, however, many engineers whose object is to work as few boilers as possible in order to reduce the radiation losses. They hold the view that if three boilers can do the work of four, one quarter of the radiation losses will be saved. This, of course, is undeniable. The author believes that any small saving by reducing the number of boilers on the range is far outweighed by the decreased efficiency that is given by a boiler on full load compared to a boiler working on 2/3 to 3/4 load. He has never had experience of a boiler, water tube or shell that was not appreciably more efficient at 20 per cent. or 30 per cent. below its rating than at full rated output.

The reason is not far to seek. If a boiler maker is asked to quote for a boiler of a certain output he will, in order to cut his price as much as possible, offer to supply the smallest boiler that he thinks will satisfy the customer, and it is quite possible that he is quite right from an economic point of view. It probably will not pay to *buy* extra boilers in order to get 4 per cent. extra efficiency, but if the boilers are already there, it will almost certainly pay to *work* them.

Let us be thoroughly angel-treading and try to work out an example. Suppose a factory wants an output of 50,000 lb. of steam an hour, and suppose that Lancashire boilers can be bought for 25*s*. per lb./hr. Seven Lancashire boilers of 7,150 lb./hr. rating will supply the demand and will cost £62,500. Suppose they have an efficiency of 70 per cent. they will burn about 6,250 lb. of coal/hour. Now suppose that 10 boilers are installed instead of 7, and that in consequence the efficiency rises to 74 per cent., the coal burnt will drop from 6,250 to 5,910 lb./hr. On a 50 hour week this will amount to a saving of 380 tons of coal a year or, with coal at 110*s*. a ton, to say £2,090. The extra boilers will have cost some £26,800 and will therefore only bring in a return of 7·8 per cent. Such a return is of course inadequate to justify the installation. If however, the factory is working three shifts the return is 23 per cent., which can well be considered sufficient. If there are many sharp peaks to meet there is little doubt that it would pay well to instal the extra boilers and have them all on about two-thirds normal rating.

If the boilers are already installed and only 7 out of 10 are at work we can save £2,090 a year on one shift and £6,250 a year on three shifts by putting them all to work apart from any saving gained by the extra ability to meet peaks.

There is an exception to this argument that it pays to work boilers, if they exist, at three-quarters to two-thirds load, and this is the case of high pressure high-brow boilers, when they supply back pressure power plant. The auxiliaries of such boilers call for a truly fantastic amount of power. Where such boilers are installed in a factory, the reason for their installation is almost certainly because the steam/power ratio is already murky. It will in such cases almost always pay to work as few boilers as possible in order to cut down the auxiliary load. It is a case of balancing the extra boiler efficiency obtained on low rating (because the high-pressure high-brow boiler, just like its humble Lancashire half-brother, is over-rated by its maker for commercial reasons) against the steam that must be condensed or blown off in order to run the extravagant feed-pumps, fans and whatnot that each extra boiler calls for.

In a power station the auxiliaries may take 4 per cent. or 5 per cent. of the power output. The plant may make 1 kWh from 8 lb. of steam. A similar industrial boiler feeding back pressure turbines may use 32 lb. steam per kWh yet will need the same auxiliaries which will now absorb 16 per cent. to 20 per cent. of the turbine output. This is a very important point that process factories sometimes overlook. A sacrifice of 1 per cent. or 2 per cent. in boiler efficiency may well yield 10 per cent. more net power. There is a process factory in England with a very beautiful high pressure boiler plant in which the boiler feed pump alone uses almost exactly 10 per cent. of the turbine output.

437. SAFETY VALVES. There is much more in a safety valve than meets the eye or strikes the ear. They can be and often are insidious, persistent steam

wasters. Their peculiarities are discussed more fully in Chapter 19. They will only be considered here in so far as they can help or hinder the storage of steam or the ability of the boilers to meet peaks.

A plain safety valve for a boiler working at 150 psi will probably require a range of 10 to 15 psi between tight shut and full open. If it is to be full open at 160 psi it will be breathing at 150 psi. The boiler house staff will become accustomed to running with breathing safety valves, because only thus are they sure that they have their boilers at full working pressure. If the safety valves are tight shut the boiler pressure is almost certainly low. So that plain safety valves have a demoralising effect and are chronic steam wasters.

The quick-lift, full-bore or pop valve has a range of only 2 to 5 psi between tight shut and full open, they open with an ear-shattering roar and need only be set to lift when the pressure is 5 psi or so too high. Such valves do not—they cannot—breathe. They give no comfortable insidious message about "a good head of steam". When they open they shout "Hi ! I'm wasting steam."

There is another common and bad safety valve habit. Many boiler operators set one boiler to blow about 5 psi before the rest. The intention is good. It is to provide a warning before all the valves lift together. But sooner rather than later it is used to deliver the comfortable words "Good head of steam".

By careful choice of safety valve design and make, by setting all the valves so that they open as nearly simultaneously as possible, by arranging the setting as high as safety will permit, by ensuring that the vent pipe is not led outside where its message is barely audible, it may well be possible to get an extra working margin of 10 psi. This may easily be the equivalent of 2 minutes 100 per cent. overload or of 2 minutes storage at zero output.

Although it is recommended above that all safety valves be set to open simultaneously, this must not be misunderstood. It means all similar safety valves. A boiler fitted with a superheater and an economiser has a safety valve on the superheater outlet, one on the boiler drum or shell, and another on the economiser. Now these safety valves must be set to lift in that order. The superheater safety valve must lift first. Were the valve on the drum to lift first when the load dropped, the superheater might get overheated. Were the economiser safety valve to lift before the drum valve the boiler might be starved of water.

All the superheater valves on all boilers should be set for one pressure. All the drum or shell valves set for a pressure slightly higher, and all the economiser valves set for a pressure appreciably higher.

438. STEAM ACCUMULATOR—RUTHS TYPE. The Ruths accumulator is shown in Fig. 253. It consists of a large steel cylindrical vessel nine-tenths filled with water. It is preferably arranged horizontally so as to give the largest possible surface of water for the liberation, as flash, of the stored steam.

One pipe A comes to the accumulator and steam either enters or leaves the accumulator through this one pipe. When the output of steam and the consumption of steam are equal there is of course no flow into or out of the accumulator. It would be silly to discharge and to charge the accumulator simultaneously ; so that one pipe is all that is necessary. The control of the accumulator is done by the two valves B and C whose action will be described later.

When steam is passed into the accumulator by the control valve B it must pass through the charging pipe D because the non-return valve E closes against it. The ingoing steam opens the charging non-return valve F and enters the charging manifold G to which are attached a number of nozzles well submerged in the water. The nozzles H, although projecting downwards, blow upwards inside the convection pipes K. Fig. 214b shows an enlarged section of a nozzle. The nozzles encourage rapid circulation, ensure quick mixing of the water in the accumulator and make certain that the steam will condense quickly and quietly without rattles and bangs.

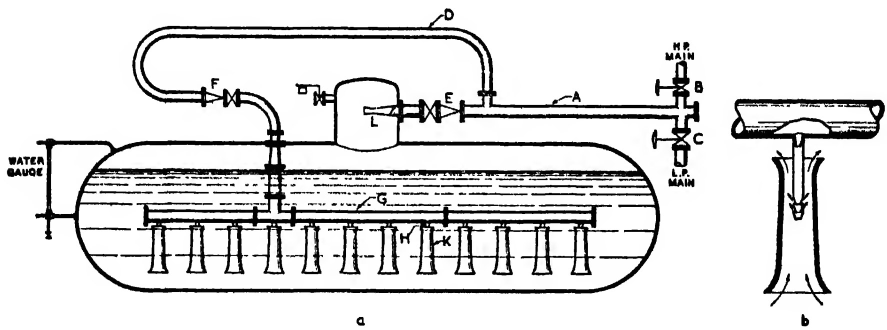

FIG. 253. RUTHS STEAM ACCUMULATOR

During charging the pressure rises in the accumulator, the water boiling point rises and so allows more steam to condense and more heat to be stored.

When the control valve C calls for a steam discharge from the accumulator the pressure in the pipe A falls below the pressure in the body of the vessel. Non-return valve F closes which prevents water being discharged, and non-return valve E opens and allows steam to escape. The lowered pressure in the accumulator causes the surplus heat in the water to be given up as flash. The nozzle L is a restriction on the flow of steam which prevents the steam discharging at a dangerously fast rate which might cause priming or carry-over. At ordinary discharge rates, the nozzle being of Venturi shape causes very little pressure drop. Any other form of restriction to the flow would cause a considerable irrecoverable pressure drop at all rates of flow.

439. STORAGE CAPACITY OF RUTHS ACCUMULATOR. The amount of steam that hot water can give up as flash has been discussed in Sections 44, 45 and 435. The water in an accumulator that is at work is always at the boiling temperature appropriate to the pressure in the vessel. This must be so, as were the water below boiling point the steam above it would condense until the vapour pressures had equalised ; were it hotter the surplus heat would cause a flash until the vapour pressures had similarly equalised.

The capacity of an accumulator for a given pressure drop is much greater at low pressures than at high pressures. A few examples will confirm this.

We will take a series of 20 psi pressure drops and work out the effects. Suppose we have a small accumulator containing 100,000 lb. of water. We can extract the following from the saturated steam table :—

Gauge Pressure	*Sensible Heat*	*Latent Heat*	*Total Heat*
0 psi	180·2	970·6	1150·8
20	227·5	940·1	1167·6
40	256·1	920·4	1176·5
60	277·1	905·3	1182·4
80	294·5	892·7	1187·2
100	309·0	881·6	1190·6

Let us see how much saturated steam at 20 psi.g. can be stored in 100,000 lb. of water at 212° F.

Let x = lb. of steam that can be stored

$$\text{Then } (100{,}000 \times 180 \cdot 2) + (x \times 1167 \cdot 6) = (100{,}000 + x)\, 227 \cdot 5$$

$$18{,}020{,}000 + 1167 \cdot 6x = 22{,}750{,}000 + 227 \cdot 5x$$

$$x = 5{,}031 \text{ lb. of steam stored.}$$

Let us now see how much steam will be given up when the accumulator is discharged from 20 psi.g. to 0 psi.g.

We have now 100,000 + 5031 = 105,031 lb. of water at 20 psi.

When the pressure is reduced to 0 psi the surplus heat available for flash is 227·5 − 180·2 = 47·3 Btu/lb.

The total steam flashed will be $\dfrac{105{,}031 \times 47 \cdot 3}{970 \cdot 6} = 5{,}118$ lb. steam at 0 psi.

Working out the other pressure intervals and starting with 100,000 lb. in water at the lower pressure boiling point we get :—

Pressure Range	*lb. Steam Charged*	*lb. Steam Discharged*
0 to 20 psi.g.	5031	5118
20 to 40	3107	3137
40 to 60	2320	2335
60 to 80	1949	1959
80 to 100	1645	1651

It will be seen that we can take out rather more steam than we put in. This is because the total heat in saturated steam at one pressure is more than the total heat in the same weight of steam at a lower pressure (below 450 psi). Radiation loss from the accumulator, though very small—see Section 446—tends to cancel this because the heat loss results in condensation, and for practical purposes we can take output from an accumulator as being the same in weight as input subject to two considerations dealt with later.

Fig. 254 shows the storage capacity of 1 cu. ft. of water over various pressure drops at various pressures. Table IV in the Appendix shows the storage capacity in terms of per cent. of flash over a large range from 250 psi to 28 in. vacuum.

When using Fig. 254 care must be taken to use the correct water density, if the figures are to be converted back to lb. water instead of cu. ft.

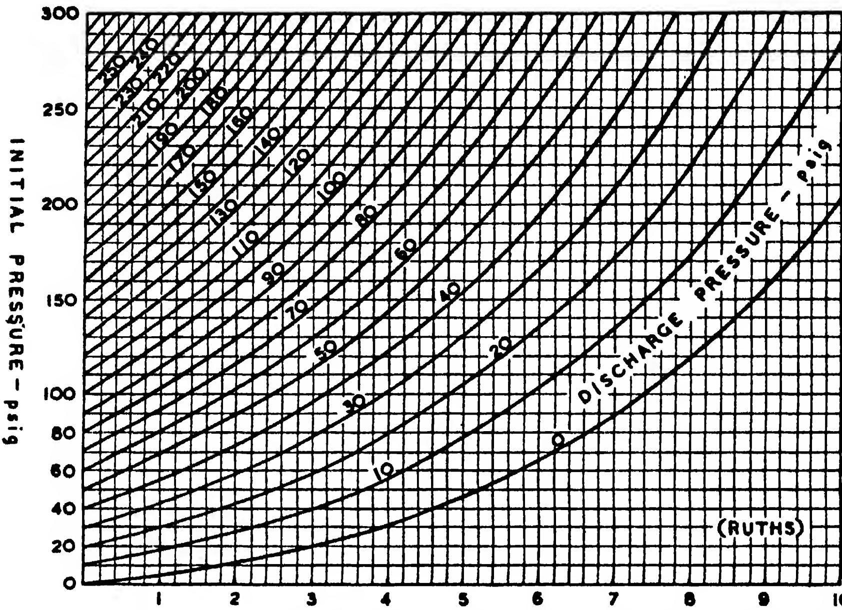

FIG. 254. STEAM STORAGE CAPACITY OF WATER

For example, flash from 80 psi.g. to 20 psi.g. gives from Fig. 254, 4·0 lb. flash steam per cu. ft. of water.

The volume of water at 80 psi.g. is found in the Steam Table to be

·0177 cu. ft./lb. Therefore the weight of water in 1 cu. ft. is $\frac{1}{\cdot 0177} = 56\cdot 5$ lb.

Table IV says that over a pressure drop from 80 psi.g. to 20 psi.g. the flash will be 7·13 per cent. Now 7·13 per cent. of 56·5 lb. is 4·03 lb., so that when correctly used Fig. 254 and Table IV agree.

Had the water density at 20 psi been taken the water weight in 1 cu. ft. would have been $\frac{1}{\cdot 0171} = 58\cdot 5$ lb. On this weight a 4 lb. flash only represents 6·84 per cent.

440. INSTANTANEOUS VERSUS GRADUAL FLASH. There is a definite difference between the flash from an accumulator from say 50 psi.g. to 0 psi.g. and the flash from condensate or blowdown from 50 psi.g. to 0 psi.g.

In an accumulator, when the pressure drops, the flash steam is given off gradually. It is not all given off at the low pressure. Let us see just what the difference is. We will take a single flash from 50 psi.g. to 0 psi.g., all the steam being flashed off at 0 psi. and an accumulator flashing in 5 psi steps from 50 psi.g. to 0 psi.g.

Weight of water at start lb.	*Pressure drop psi.g.*	*Sensible heat*			*Latent heat at lower pressure*	*Weight of flashed steam lb.*
		Upper	*Lower*	*Difference*		
1,000	50— 0	267·4	180·2	87·2	970·6	89·84
1,000·000	50—45	267·4	261·9	5·5	916·2	6·003
993·997	45—40	261·9	256·1	5·8	920·4	6·264
987·733	40—35	256·1	249·8	6·3	924·9	6·728
981·005	35—30	249·8	243·0	6·8	929·7	7·175
973·830	30—25	243·0	235·8	7·2	934·6	7·502
966·328	25—20	235·8	227·5	8·3	940·1	8·532
957·796	20—15	227·5	218·4	9·1	946·0	9·212
948·584	15—10	218·4	207·9	10·5	952·9	10·452
938·132	10— 5	207·9	195·5	12·4	960·8	12·107
926·025	5— 0	195·5	180·2	15·3	970·6	14·597
						88·572

If the steam from the accumulator feeds a main at 0 psi.g. the steam emerging into the L.P. main will be wiredrawn and slightly superheated. The amount of energy liberated by the accumulator must be the same whether done in instalments or in one sudden drop. Therefore, if the 88·57 lb. of steam given off wiredrawn at 0 psi in instalments of 5 psi were desuperheated so that the whole output were saturated at 0 psi.g., the desuperheated steam would weigh 89·84 lb.

The difference between the gradual flash and the one step flash is 89 84 — 88·57 = 1·27 or 1·4 per cent. The calculation can be done as a straight single flash provided the loss of sensible heat of the water is less than 30 Btu/lb. If a single flash results in a greater sensible heat drop than 30 Btu, the calculation should be done by steps so that each step is below a 30 Btu drop of sensible heat. If this is done the error will never exceed ¼ per cent.

441. WATER LEVEL VARIATIONS IN ACCUMULATOR. Now we have seen that when 5,000 lb. of steam is blown into 100,000 lb. of water the resulting 105,000 lb. of water will give up about 5,100 lb. of lower pressure steam. Unless condensation due to heat loss is sufficient to make up for this larger discharge there will be a small water loss each time the accumulator is discharged. If the charging steam is superheated the amount of water lost at each discharge will be greater. Here are some examples.

The amount of charging steam is found as in the previous section, where x is the weight of steam.

Heat in original water + (x × Total heat in charging steam/lb.) = (x + Original water) × Sensible heat at the charged pressure.

Pressure Range psi.g.	*Charging Steam Superheat*	*Quantity Put in lb.*	*Quantity Discharged lb.*	*Water Loss per Discharge lb.*	*per cent.*
0 to 20 to 0	*Sat.*	5031	5118	87	·087
0 to 20 to 0	50° F.	4897	5112	215	·215
0 to 20 to 0	100° F.	4777	5106	329	·329
0 to 20 to 0	200° F.	4572	5096	524	·524

If the steam that is charged into the accumulator is wet, the contrary effect takes place ; the water in the accumulator increases. We can find out the amount of the charging steam by the same method, except that the heat in the charging steam will be :—

Sensible heat in charging steam + (Latent heat × dryness fraction).

Working out some examples we get :—

Pressure Range	*Charging Steam Wetness*	*Quantity Put in lb.*	*Quantity Discharged lb.*	*Water gain or loss per Discharge*
0 to 20 to 0	Dry	5031	5118	87 loss
	per cent.			
0 to 20 to 0	5 wet	5296	5131	165 gain
0 to 20 to 0	10 wet	5590	5146	444 gain
0 to 20 to 0	15 wet	5919	5162	757 gain
0 to 20 to 0	20 wet	6289	5180	1109 gain

As the conditions must vary and as the quality of the steam may not be known, it is necessary to keep an eye on the accumulator water gauge and to make provision for blowing down excess water or for feeding in any lost water needed to maintain the level. When appreciable water has to be blown down, as happens in some accumulator applications, as will be seen later, provision should be made for collecting any flash and for collecting the water, which is clean pure condensate.

442. RATE OF DISCHARGE FROM ACCUMULATORS. The rate at which an accumulator can be allowed to discharge is limited by the rate at which ebullition can take place at the liquid surface without the entrainment of water droplets. Fig. 255 shows the maximum rate of discharge recommended by the makers of the Ruths accumulator, based on their wide experience. This is a straight line, which, if continued to the left, would cut the pressure ordinate at approximately zero pressure absolute. If we take points on the curve we get the following :—

18 psi.g. or 33 psi.a. the permissible rate of discharge is .. .. 100 lb./sq. ft./hr.

52 psi.g. or 67 psi.a. the permissible rate of discharge is .. .. 200 lb./sq. ft./hr.

86 psi.g. or 101 psi.a. the permissible rate of discharge is .. .. 300 lb./sq. ft./hr.

Quite obviously a simple rule that is quite near enough for all practical purposes is

Max. rate of discharge in lb./sq. ft./hour = 3 × psi.a.

This is a very easy figure to carry in the head. Let us see whether this discharge limitation imposes serious difficulties on design.

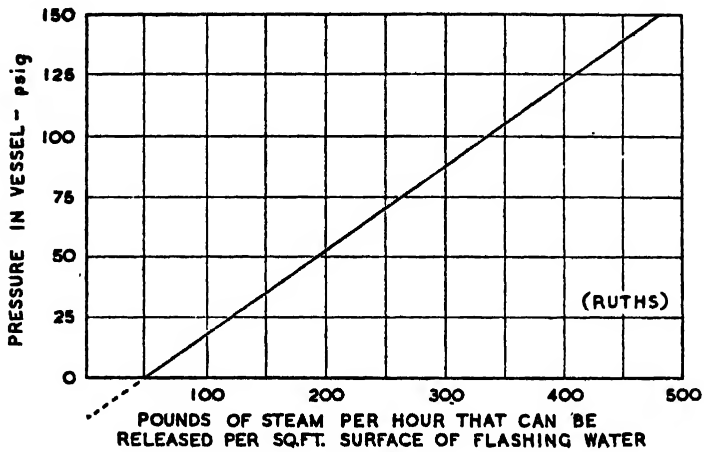

FIG. 255. PERMISSIBLE RATE OF FLASH WITHOUT ENTRAINMENT FROM A WATER SURFACE

443. **DESIGN OF RUTHS TYPE ACCUMULATORS.** In the two previous Sections we have been considering a small accumulator containing 100,000 lb. of water and capable of storing some 5,000 lb. of steam between 20 and 0 psi.g. Tables LX, LXI and Fig. 256 will help us to look at the problems, which are very simple. We will look upon the accumulator as being a plain cylinder. (In practice the ends are almost always hemispherical. The correction for this can be made afterwards. The volume of a sphere is equal to that of the same diametered cylinder 2/3 of a diameter in length).

Table LXI and Fig. 256 tell us that if the accumulator is 90 per cent. full, the water will stand 84 per cent. up the vessel and the water surface will be 73 per cent. of the surface at the diameter.

Using Fig. 256, we know the vessel is 90 per cent. full, therefore we start at A where 90 per cent. of the capacity is the liquid volume. We run vertically up the 90 per cent. line until it cuts the volume curve at B. This shows that the depth of the liquid will be 84 per cent. up the vessel. The 84 per cent. depth line cuts the liquid surface curve at C showing the surface to be 73 per cent. of that at the diameter.

Table LX tells us that a vessel 10 feet in diameter contains 489·2 gallons per foot of length. As the accumulator will be only 90 per cent. full the capacity will be 440 gallons per foot.

TABLE LX. CAPACITY OF CYLINDRICAL TANKS OF VARIOUS DIAMETERS

Gallons per Foot of Length.

DIAMETER FEET	0	1-in.	2-in.	3-in.	4-in.	5-in.	6-in.	7-in.	8-in.	9-in.	10-in.	11-in.
0	—	·0340	·1359	·3058	·5436	·8493	1·223	1·665	2·174	2·752	3·397	4·111
1	4·892	5·741	6·659	7·644	8·697	9·818	10·07	12·26	13·59	14·98	16·44	17·97
2	19·57	21·23	22·97	24·77	26·63	28·57	30·58	32·65	34·79	37·00	39·27	41·62
3	44·03	46·51	49·06	51·67	54·35	57·11	59·93	62·82	65·77	68·79	71·89	75·04
4	78·28	81·57	84·93	88·36	91·86	95·43	99·06	102·8	106·5	110·4	114·3	118·3
5	122·3	126·4	130·6	134·8	139·2	143·5	148·0	152·5	157·1	161·7	166·5	171·3
6	176·1	181·0	186·0	191·1	196·2	201·4	206·7	212·0	217·4	222·9	228·4	234·0
7	239·7	245·5	251·3	257·1	263·1	269·1	275·2	281·3	287·5	293·8	300·2	306·6
8	313·1	319·6	326·3	333·0	339·7	396·6	353·5	360·4	367·5	374·5	381·7	389·0
9	396·3	403·6	411·1	418·6	426·2	433·8	441·5	449·3	457·1	465·0	473·0	481·1
10	489·2	497·4	505·7	514·0	522·4	530·8	539·4	548·0	556·6	565·4	574·1	583·0
11	592·0	600·9	610·0	619·2	628·4	637·6	647·0	656·4	665·9	675·4	685·0	694·7
12	694·5	714·3	724·2	734·1	744·1	754·1	764·4	774·6	784·9	795·2	805·7	816·2
13	826·7	837·4	848·1	858·9	869·7	880·6	891·6	902·6	913·7	924·9	936·1	947·5
14	958·8	970·3	981·8	993·4	1,005	1,017	1,029	1,040	1,052	1,064	1,076	1,089
15	1,101	1,113	1,125	1,138	1,150	1,163	1,175	1,188	1,201	1,214	1,226	1,239
16	1,252	1,265	1,279	1,292	1,305	1,318	1,332	1,345	1,359	1,373	1,386	1,400
17	1,414	1,428	1,442	1,456	1,470	1,484	1,498	1,513	1,527	1,541	1,556	1,570
18	1,585	1,600	1,615	1,629	1,644	1,659	1,674	1,689	1,705	1,720	1,735	1,751
19	1,766	1,782	1,797	1,813	1,829	1,844	1,860	1,876	1,892	1,908	1,924	1,941
20	1,957	1,973	1,990	2,006	2,023	2,039	2,056	2,073	2,089	2,106	2,123	2,140

We should therefore need a vessel $\frac{10,000}{440} = 22 \cdot 72$ feet long.

The liquid surface will be 73 per cent. of $22 \cdot 72 \times 10 = 165 \cdot 9$ sq. ft.

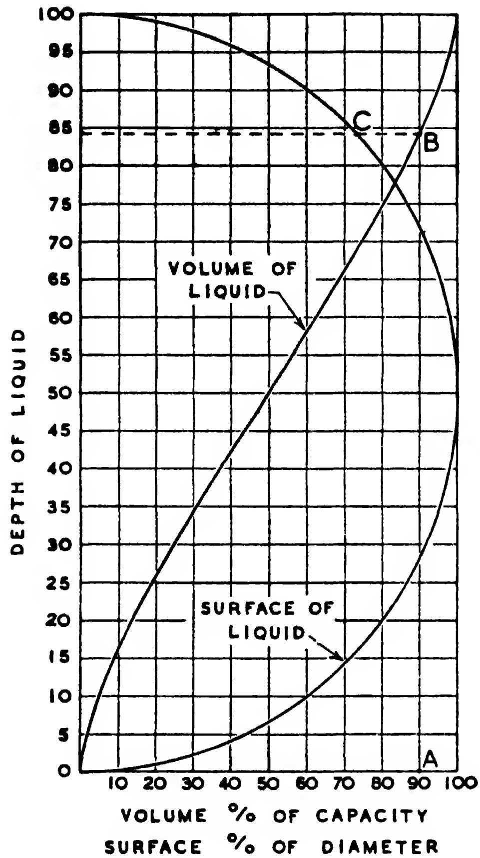

FIG. 256. CAPACITY AND LIQUID SURFACE OF PARTLY FILLED HORIZONTAL CYLINDRICAL TANKS

Fig. 255 tells us that at atmospheric pressure the water surface can be allowed to flash off 50 lb./sq. ft./hour. So that the discharge rate of our proposed accumulator will be $50 \times 165 \cdot 9 = 8,295$ lb./hour.

If we wish to use the full capacity, 5,000 lb., of this accumulator we can only allow it to discharge its 5,000 lb. over a period of 36 minutes. Clearly such an accumulator will be of little use for dealing with short-term peaks. It would however be very useful for storing steam in the morning for use in the afternoon.

Let us try a longer, smaller diameter vessel. If it is 7 ft. 6 in. in diameter, Table LX tells us that each foot will hold 275·2 gallons. 90 per cent. of 275·2 is 247·7. Its length will therefore be $\frac{10,000}{247 \cdot 7} = 40.4$ feet.

The liquid surface will be 73 per cent. of $40 \cdot 4 \times 7 \cdot 5 = 221 \cdot 2$ sq. ft.

The safe discharge rate will be $50 \times 221 \cdot 2 = 11,060$ lb./hour.

Such a vessel will have a minimum discharge time of 27 minutes for 5,000 lb. of steam. This does not show a very greatly improved discharge rate over the 10 ft. vessel.

TABLE LXI. LIQUID SURFACE AND CONTENTS OF HORIZONTAL CYLINDRICAL TANKS

PER CENT. DEPTH	CONTENTS PER CENT. OF VOLUME	SURFACE PER CENT. OF CROSS SECTION
5	1·87	43·7
10	5·20	59·9
15	9·41	71·2
20	14·23	80·0
25	19·55	86·7
30	25·23	91·6
35	31·19	95·4
40	37·36	98·1
45	43·64	99·6
50	50·00	100·0
55	56·36	99·6
60	62·64	98·1
65	68·81	95·4
70	74·77	91·6
75	80·45	86·7
80	85·77	80·0
85	90·59	71·2
90	94·80	59·9
95	98·13	43·7
100	100·00	0

Before discussing how this difficulty can be overcome, let us look for a moment at higher pressure conditions.

Let us take an accumulator working between 40 and 20 psi.g. In Section 439 we saw that the storage within this range was 3,100 lb. in 10,000 gallons.

The permissible rate of discharge at 20 psi is

$(20 + 15)\ 3 \times 165 \cdot 9 = 17,420$ lb./hr. if the vessel is 10 ft. $\times$ 22·72 ft.

This gives a discharge time of 11 minutes for 3,100 lb.

If the dimensions are suitably increased to provide storage for 5,000 lb. the discharge time will be about 18 minutes. By making the accumulator long and thin the discharge rate can be increased to about 12 minutes—quite a useful increase.

If we work out conditions for higher pressures, we find that the higher the pressure the quicker can be the discharge. So it is only at low pressures that the limitation due to insurance against carry-over is really irksome.

The reason for keeping the discharge rate low is twofold. At high pressures carry-over might cause a water hammer which would be more dangerous than at low pressure. The high pressure accumulator is very costly, and the smaller it can be made the cheaper it will be. It is therefore filled as full as possible so as to increase its capacity compared to its volume. If the water level is lowered the accumulator must be bigger for a given water content; this raises its cost. The larger a vessel is for a particular pressure the thicker must be the plates, which again raises the cost.

At low pressures the plates can be quite thin, and a large increase in volume can be obtained relatively cheaply. For low pressure working therefore accumulators are made larger and are not filled so full. An accumulator of fair diameter filled just over half can be discharged at twice the rate of a normal 90 per cent. filled accumulator.

Suppose we make the accumulator 12 feet in diameter and that we fill it 60 per cent. in depth. Table LXI tells us that at 60 per cent. full it will contain 62·64 of its volume. Table LX tells us that the content will be 694·5 gallons per foot length. Its accumulator capacity will be 694·5 × ·6264 = 435 gallons per foot. To hold 10,000 gallons will call for a length of 23 ft. The surface, from Table LXI, will be 23 ft. × 12 × ·981 = 271 sq. ft. If we wish to discharge 5,000 lb. each sq. ft. will have to evolve 18·5 lb. If the permissible rate is now double, or 100 lb./sq. ft./hr. we can get full discharge in just over 11 minutes. It only needs a relatively small increase in dimensions to get any rate that is within reasonable bounds.

The Ruths type of accumulator is seldom used at very low pressures round about atmospheric, but it has been shown that the design of such accumulators presents no difficulty if the level is lowered and the discharge rate increased. Above about 20 psi the psi.a. × 3 rule gives an accumulator that is quite reasonable in size and shape for most ordinary discharge rates and capacities even if it is 90 per cent. full of water.

444. CONTROL OF RUTHS TYPE ACCUMULATOR. The Ruths accumulator is a cushion between the steam producer and the steam user, generally between the boilers and a process load. It is possible to use a Ruths accumulator for dealing with peak power loads, but a considerable forfeit has to be paid in the form of pressure drop from which no power is got. The chief application for an accumulator of this type is for buffering the peaks that arise from low or medium pressure processes.

The boilers can be fired at a steady average rate. While the process demand is at the steady average, the boilers feed the process in a straightforward way. If part of the process plant is suddenly shut down the boilers continue to produce steam at the average rate and the excess steam charges the accumulator. When the process makes a sudden call for steam the accumulator discharges with the boilers still working steadily.

Fig. 257a shows a simple back pressure plant where the boilers supply an engine with steam at 150 psi. The engine exhausts at 25 psi into the process main. When the process calls for more steam than can be supplied by the engine exhaust the shortage is made up through the reducing valve. When the process takes less steam than is being exhausted by the engine the surplus blows away through the safety valve on the 25 psi main. If the engine is on a fairly steady load and the process demand fluctuates all the peaks and valleys must be taken by the boilers and there will be considerable pressure variations on the 150 psi main. This has a bad effect on the engine performance, is very bad for good boiler operation and wastes steam. The dotted line shows the pressure impulse that actuates the reducing valve.

Fig. 257b shows an accumulator fitted between the 150 psi main and the 25 psi main. The boilers can now be fired at a steady rate, the engine gets a steady pressure, any sudden demand is met by the accumulator and any excess boiler production charges the accumulator. When the boiler pressure rises to or above normal the surplus valve S opens, under impulse from the 150 psi main. The reducing valve, R, impulsed from the 25 psi main demands steam from the accumulator main whenever the 25 psi pressure drops.

This system clearly has limitations. The 25 psi main might call for steam when little was being surplussed and when the accumulator was empty. There would then be a shortage of steam for the process and the fireman would not be aware of the situation. There must be some way of communicating the process demand to the high pressure main. The way this is done is shown in Fig. 257c. Another reducing valve is fitted between the high pressure main and the accumulator line. This valve is impulsed from the accumulator main. This system meets all the requirements.

Now, as the process pressure is considerably lower than the boiler pressure, there may be no need to use the full pressure drop over the accumulator which could then be made more cheaply to stand a somewhat lower pressure. Suppose we say that the accumulator is to work between 100 psi and 25 psi. From Table IV we see that the flash from 150 to 25 psi is 11·0 per cent. while from 100 to 25 psi the flash is only 7·83 per cent. This is a reduction in storage capacity of 29 per cent. It may be that a somewhat larger accumulator at the lower pressure will be cheaper than the high pressure smaller vessel, or it may be that the smaller capacity may prove adequate. Assuming that the lower pressure is decided on—this arrangement is quite common in practice—what difference must be made to the controls ?

The accumulator must be protected from the boiler pressure otherwise as soon as its pressure was up to 100 psi its safety valve would blow. A master valve is, therefore, fitted between the accumulator supply valves and the accumulator line. This valve is impulsed by the accumulator pressure and closes as soon as the accumulator is full, that is as soon as its pressure reaches 100 psi. The arrangement is shown in Fig. 257d. The reducing valve R_1 ensures that the low pressure main is supplied with steam. The reducing valve R_2 transmits the request for process steam to the boilers when the accumulator is empty and the surplus valve is shut. The surplus valve allows the boilers to generate steam when the demand is small and the surplus is then stored in the accumulator. The master valve protects the accumulator from excessive pressure and prevents steam being wasted by the accumulator safety valve.

In practice the three valves that control the steam into the accumulator are all combined into one valve that receives and acts on the three impulses. This is shown in Fig. 257e. The impulses are shown in the diagrams as dotted lines. The action of this valve is described in Chapter 19.

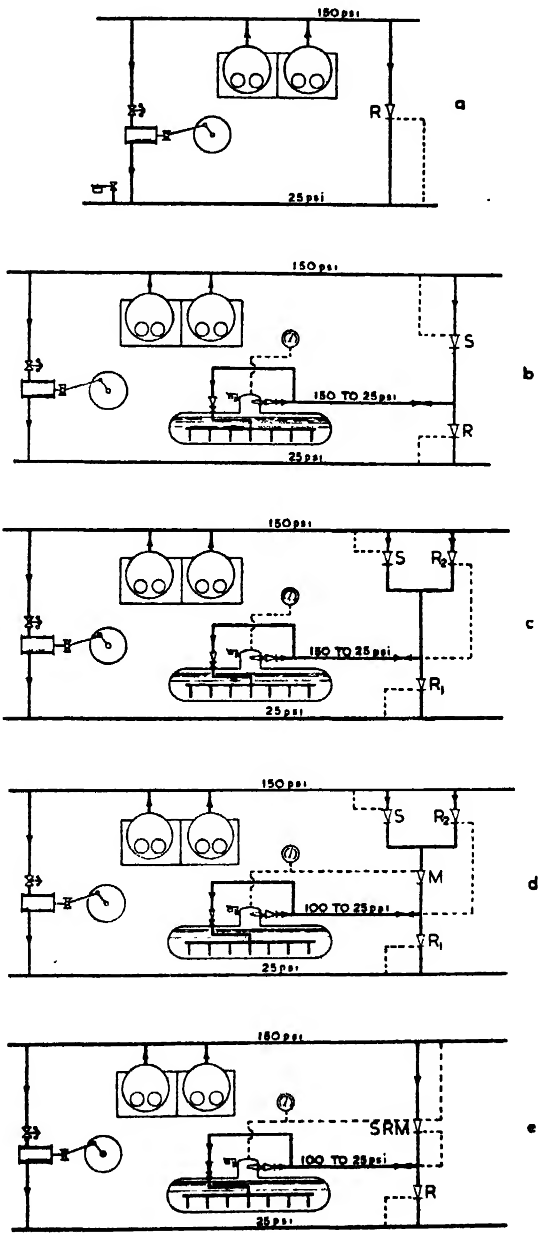

FIG. 257. DEVELOPMENT OF THE CONTROL OF A RUTHS ACCUMULATOR

445. BOILER FIRING GAUGE. Attached to the accumulator is a pressure pipe leading to a special pressure gauge fitted in some prominent position on the boiler firing floor. The essence of the accumulator is that it permits the boilers to operate at continuous load. This continuous load should be the average load. But the average load will vary not only from day to day but to some extent throughout the day. There must be some method of informing the boiler house whether the average load is being met. This special pressure gauge delivers this message. We will assume that the accumulator is arranged to work over the whole range between boiler pressure and process pressure as in Fig. 257c. The accumulator firing gauge will then look like Fig. 258. This gauge shows

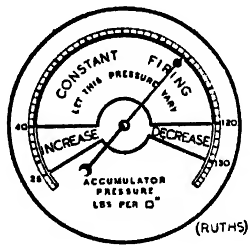

FIG. 258. RUTHS BOILER FIRING GAUGE

the pressure inside the accumulator. If the accumulator pressure gets very low the fireman is instructed to increase his rate of firing. If the accumulator pressure then rises he maintains the new rate of firing throughout all the accumulator pressure variations between 40 psi and 120 psi. He will probably take no notice of the first instruction to increase his firing rate, but if the gauge persistently gives an "increase" message he will make the change. Similarly a persistent call to reduce his firing rate will be obeyed and a drop in accumulator pressure will be ignored until the gauge persistently urges him to produce more steam again. By means of this gauge he is enabled to adjust the average rate of firing to suit the true temporary average demand and the accumulator is enabled to perform its full function of levelling out the peaks and valleys.

446. LOSS OF HEAT FROM ACCUMULATORS. An accumulator is usually very well lagged with 4 in. or 5 in. of first class lagging. It then loses heat very slowly. Fig. 259 shows the loss of heat shown by a test taken on the accumulator in the author's factory in 1933 over a period of eight days during which no steam was taken from or put into the accumulator. It shows that the heat loss was such that the pressure dropped by 10 psi per day. The heat loss amounts to the remarkably low figure of ·175 Btu/sq. ft./° F./hr. Normally, of course, the accumulator is only intended to store steam for a few hours at most, but Fig. 259 shows how useful it can be over extended periods.

In a sugar refinery, for instance, at the shut-down on Saturday the power load persists for several hours longer than the process steam load. It is very convenient to be able to store the resulting exhaust steam that would otherwise have to be wasted. After a few weeks practice the boiler house staff are able

to arrange things so that the accumulator is empty just before the shut-down starts. Steam is then available over the week-end for the canteen, etc., and for helping to warm up the factory on Sunday night.

Before discussing the applications of the Ruths accumulator we will look at the other types of accumulator. We can then see how each type has its own peculiar qualities making it particularly suitable for the various situations which can arise in different plants.

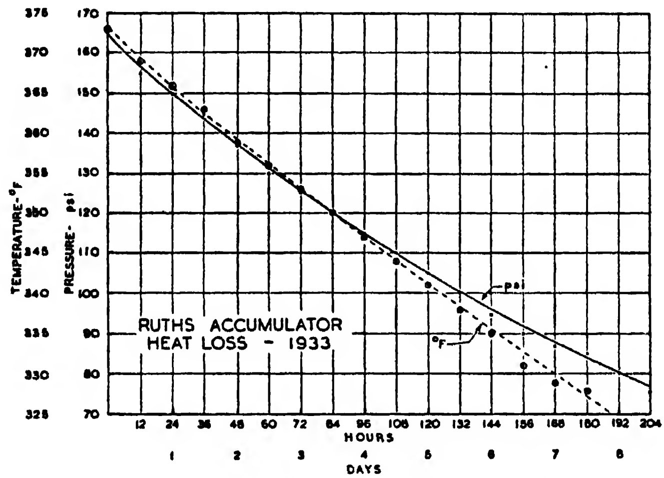

FIG. 259. HEAT LOSS FROM ACCUMULATOR OVER EIGHT AND A HALF DAYS

447. EXHAUST ACCUMULATORS—RATEAU TYPE. The Rateau accumulator was the first accumulator to be made and used. It works on exactly the same principle as the Ruths accumulator but it operates over a much smaller pressure range and the control is very much simpler—in fact there are virtually no controls.

A Ruths accumulator is generally used to smooth fairly long-term *demand* peaks. A Rateau accumulator is fitted to absorb very short sharp *supply* peaks. The qualities that are essential for a Rateau exhaust accumulator are that it must be able to absorb and condense very sudden very large short puffs of steam, that it must then discharge the stored steam evenly and relatively slowly and that it must offer the smallest possible resistance to the steam flow and must operate over the smallest possible pressure drop.

To absorb steam quickly there must be a great number of nozzles and the water circulation must be very quick, certain and complete. To discharge evenly and quickly is easy, but as such an accumulator is concerned with minutes and seconds, not with hours, the water surface must be particularly large. To minimise pressure drop the nozzles should project as little as possible below the surface, but unless there is a definite submergence and very good circulation the steam will break the surface and only the top layer of water will be hot and will act as accumulating medium.

448. TYPES OF EXHAUST LOAD TO BE STORED. Fig. 260 shows the exhaust steam produced by a winding engine operating at 40 winds per hour. During the first 6 to 8 seconds acceleration takes place at full admission. The cut-off is then set at about 30 per cent. and the wind is finished after 30 seconds. The engine is then stopped for one minute.

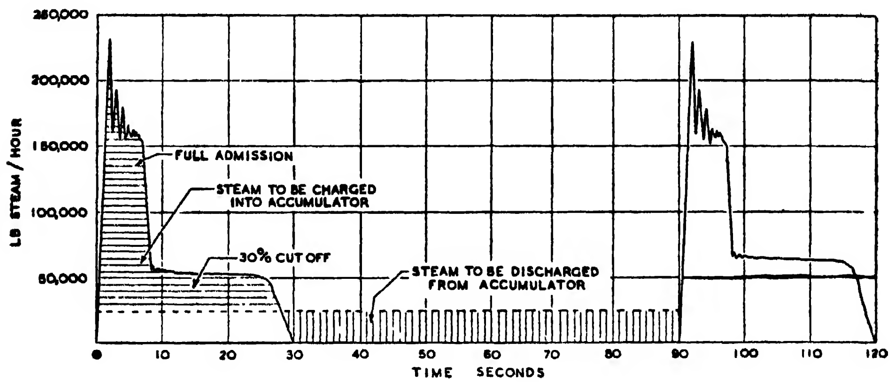

FIG. 260. STORAGE OF WINDING ENGINE EXHAUST

The exhaust steam is used in an exhaust turbine on a steady load. The steam used per wind is about 650 lb. spread over 30 seconds, of which about 300 lb. is exhausted in the first 8 seconds. The average steam consumption is $650 \times 40 = 26{,}000$ lb./hour, or about 220 lb. per half minute which is the steaming period of each wind. The surplus steam that must be stored during the steaming of the engine is $650 - 220 = 430$ lb. This steam must be stored in 30 seconds and given up during the following 60 seconds. So that the rate of charging must be double the rate of discharge.

The discharge rate when the engine is stopped is 26,000 lb./hour, so that, with an emission rate of 100 lb./sq. ft./hr. the water surface must be 260 sq. ft. or say 300 to provide an ample margin. If the accumulator is about half full its dimensions must be about 10 ft. × 30 ft.

If the accumulator is 10 ft. × 30 ft. and is to be 60 per cent. full, it will have a water content (see Table LX) of $489{\cdot}2 \times {\cdot}6 \times 30 = 8{,}806$ gallons or 88,060 lb. We have to store 430 lb. of steam or ·49 per cent. of the water content. Table IV shows that a pressure drop from 1·5 psi.g. to atmospheric pressure will just cover this theoretically. But things are not quite as good as this. We must allow for some small pressure and temperature drop to ensure quick condensation—say ·5 psi. There is also the hydrostatic head due to the submergence of the nozzles. If they are just over one foot submerged this adds another ·5 psi.

So that we can say that a 10 ft. × 30 ft. accumulator will store the exhaust from the winder with an output pressure of 0 psi.g. and an input pressure or engine exhaust back pressure of 2·5 psi.g.

Rolling mill engines and hammers present similar but more irregular characteristics.

449. WATER LEVEL IN EXHAUST ACCUMULATORS. In Section 441 it has been explained that if dry saturated steam is charged into an accumulator each cycle of charge and discharge will cause a slight loss of water. If, however, wet steam is blown in, the wetness adds itself to the water in the accumulator, and, as the water loss with saturated steam is largely balanced by heat loss condensation, the water level will rise by the exact amount of the wetness.

An exhaust accumulator is almost always used to store the exhaust from reciprocating engines and such exhaust is almost always wet, sometimes very wet indeed. Provision must be made to draw the water off and is easily done by means of a weir and an ordinary trap.

If the exhaust steam is superheated, a possibility, but unlikely, the accumulator will lose water and provision must be made for pumping water in. This can readily be done by a connection from the boiler feed pump and can be self-adjusting by means of a float valve or its equivalent or can be done regularly by hand to a mark on the water gauge glass.

450. EFFECT OF OIL. The exhaust steam from a reciprocating engine may contain oil. It will certainly contain oil if the steam was superheated. However hard we try to separate the oil from the exhaust steam there will always be a gradual accumulation of oil on the surface of the water. The rupturing of this oil film, or oil layer as it may well become, offers a serious resistance to the flashing steam when the accumulator is discharging. Quite a thin layer of oil may easily double the pressure drop needed to release a given amount of steam. Any increase in pressure drop is so much dead loss of power available from the exhaust turbine. Every possible effort must therefore be taken to keep the water surface as free from oil as possible.

Exhaust steam from a lubricated reciprocating engine should always pass through an oil separator before entering an accumulator.

Oil removal in an exhaust accumulator is done by means of skimming devices, a plain weir being effective provided there is a current on the surface of the water towards the weir. This is often secured by designing the nozzles to induce the circulation currents at the surface to travel towards the weir. A trap on the overflow side of the weir will generally discharge the oil together with the surplus water.

If the accumulator is taking superheated steam and has in consequence to be fed with water to maintain the working level, or if the steam is saturated or only slightly wet so that no water would normally be added, it is necessary to add surplus water to float the oil off the surface.

451. DESIGN OF EXHAUST ACCUMULATORS. The exhaust accumulator is just the same basically as the Ruths accumulator, except that it is a cheap thin-shelled vessel and requires as a rule no controls. The most important part is the nozzle arrangement because the first essential of an exhaust accumulator is to absorb instantly and completely sudden and heavy exhaust puffs.

Fig. 261 shows a design of Rateau accumulator. The nozzles, one of which is shown enlarged in Fig. 261b, are somewhat similar to the Ruths nozzles but the actual steam discharge openings are narrow slits so as to break the steam up into the smallest possible streams. The convection tubes surrounding the

nozzles are closed at their tops and provided with side openings, all the openings pointing one way so as to induce a water flow at the surface towards the oil skimming weir at the right hand end of the accumulator. The actual steam opening of the nozzles are about 9 in. below the water surface giving an hydrostatic head of only $\frac{1}{3}$ psi.

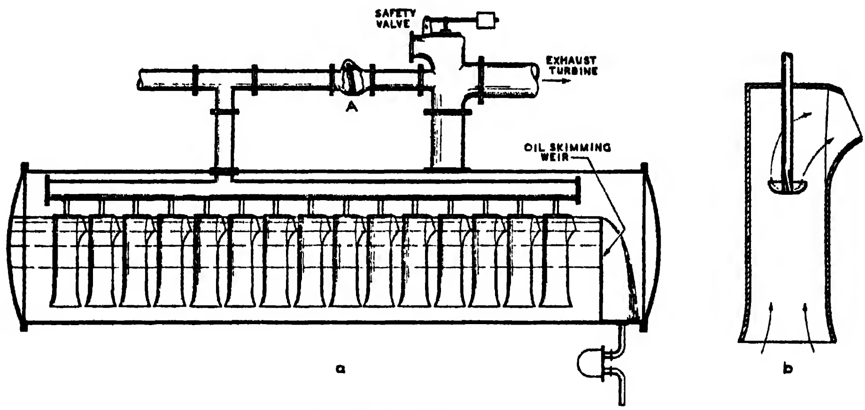

FIG. 261. RATEAU EXHAUST ACCUMULATOR

A valve A, either a piston operated butterfly valve or a simple flap valve as shown, is usually fitted so that all the exhaust steam does not pass through the accumulator. This opens when the exhaust turbine demand is heavy, and closes when the demand falls off, thus forcing the steam to go through the nozzles.

To drain away the water due to the wetness of the steam a weir is fitted at one end and a trap discharges the excess water together with the oil.

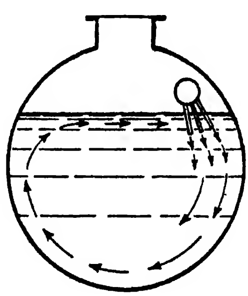

FIG. 262. TANGENTIAL INJECTION EXHAUST ACCUMULATOR

Fig. 262 shows another design in which the nozzles are put at one side of the accumulator and point downwards. They are only about 4 in. below the surface. To prevent steam breaking the surface they rely on putting the whole water contents into rapid rotational circulation. The nozzles are just plain

lengths of 1 in. pipe. In an accumulator such as was discussed in Section 447, 10 ft. × 30 ft. and storing 430 lb. in 30 seconds, there would have to be nearly 400 such nozzles.

Fig. 263 shows a circular accumulator made of reinforced concrete. Such accumulators need very little maintenance and can be made very large cheaply, while their construction does not encourage heat loss. This type of accumulator does not lend itself to easy oil skimming.

452. EXHAUST ACCUMULATORS BELOW ATMOSPHERIC PRESSURE. There is no reason whatsoever why an exhaust accumulator should not work under vacuum. The construction of the accumulator is identical with that of an accumulator working at above atmospheric pressure except that the shell must be a little stiffer to resist collapse. The water cannot rise in the steam pipes because there will be no more pressure drop than with a pressure accumulator, and it is pressure drop that causes water to stand up the pipes.

Air will of course tend to leak in, but a little air leakage does not matter very much as it will be carried out with the steam and removed by the vacuum pump attached to the condenser of the exhaust turbine.

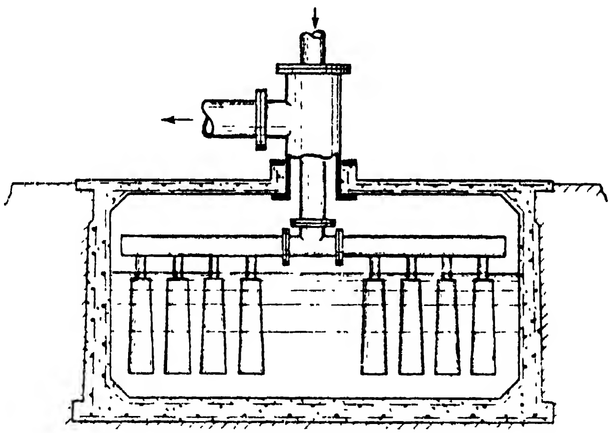

FIG. 263. CONCRETE EXHAUST ACCUMULATOR

453. FEED WATER ACCUMULATORS. The Rateau accumulator is designed to absorb the exhaust from an intermittently running engine. It does nothing towards smoothing out the sudden demand for steam from the boilers except that, in conjunction with a mixed pressure turbine it lessens the live steam demand on the boilers. We have seen in Section 435 that an increased water capacity in the boilers goes a very long way towards meeting sudden sharp demands. We have also seen that by having an ample boiler capacity the boiler house is in a better position to meet sudden draws. During a peak demand everything possible should be done to help the boiler and to relieve its load. Similarly, extra load should, if possible, be found for it during the valley periods. This is what the feed water accumulators try to do. When

the plant is lightly loaded feed water is heated to boiling point and stored. When the peak comes along feed heating stops and the large body of hot feed can also act as a straight accumulator and give up steam by flash.

There are two kinds of feed water accumulator, the constant temperature varying volume type, and the varying temperature constant volume type. Both work at constant or virtually constant pressure. In the constant temperature or Kiesselbach design water is fed into the system in inverse proportion to the load. Actually it is done by feeding in direct proportion to the boiler pressure. During times of light load the accumulator fills up with water at boiling temperature. When the peak demand develops the feed is shut off and the boiler draws from the accumulator.

In the constant volume or Marguerre design the accumulator is made so that the hot water can collect at the top without mixing with cool water at the bottom. The accumulator is kept always full. During times of light load the incoming feed and cool water from the bottom of the vessel is heated and put into the top of the accumulator. During periods of peak the cool incoming feed is passed cool into the bottom of the accumulator. The Kiesselbach design is more suitable for short sharp demands, because it lends itself to acting as a flash vessel. The Marguerre is more suitable for long-term peaks and has the advantage that there are not such wide fluctuations in the feed water demand, thereby helping the water treatment plant. Both these accumulators are essentially aids to the boiler rather than aids to the process, though of course good boiler work makes for good process work. Feed water accumulators operate by changes in the boiler pressure, but are arranged so that these boiler pressure fluctuations are very small, while the introduction of the accumulator reduces the pressure fluctuations still more. They are therefore the ideal type of accumulator for use with power plant as distinct from process plant.

454. THE KIESSELBACH ACCUMULATOR. Fig. 264 shows the Kiesselbach constant temperature variable volume accumulator applied to an installation of small water tube boilers. The feed pump feeds the boiler through the automatic feed regulator. This regulator is impulsed from the boiler steam pressure. When the pressure drops slightly the feed is reduced, and if the pressure drops further the feed is cut off altogether. When the pressure rises the regulator opens until at a few psi above normal boiler pressure the regulator is full open. Over the pressure range which it is desired to maintain, say 195 to 205 psi, the feed varies directly with the pressure.

Circulating pipes are fitted at either end of the boiler drums at the water level. These circulating pipes lead to the accumulator and from the circulating pump. The circulating pump draws water from the accumulator and pumps it into the boiler ; the surplus water above normal water level runs out of pipe B by gravity back to the accumulator. Pressure is equalised between the boiler and the accumulator by means of the balance pipe A. The water in the accumulator is thus always maintained at boiling temperature. When the boiler is evaporating less water than the feed, that is when the load is light and the pressure is high, the accumulator fills up. When the boiler is called upon for extra steam, the pressure drops slightly, the feed stops and the water level in the accumulator drops. At the same time the reduction in pressure on the accumulator water causes a flash of steam from the accumulator water.

The regulator must be a high grade machine. It must respond to very small pressure changes and work smoothly. The control of the accumulator depends entirely on the regulator and on the rate of firing. The rate of firing can remain constant provided the water is always in the middle gauge glass on the accumulator. If the water level falls persistently into the lower glass the rate of firing should be increased. If the water level is persistently rising into the top glass it is an indication that the rate of firing should be reduced.

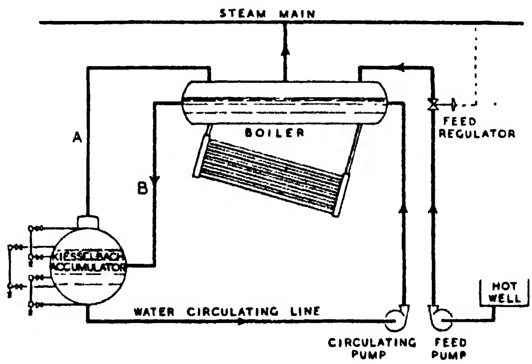

FIG. 264. KIESSELBACH FEED WATER ACCUMULATOR

455. **THE MARGUERRE ACCUMULATOR.** The Marguerre accumulator works, like the Kiesselbach, at virtually constant pressure, but the temperature of the water in the accumulator varies and there is always a constant quantity of water in the accumulator.

Fig. 265 shows the Marguerre accumulator and its application to a small water-tube boiler. The accumulator consists of a large vertical cylinder at boiler pressure. It is kept full by the primary feed pump whose output is controlled by the regulator A shown diagrammatically as being impulsed by a ball float in the top of the accumulator. The boiler draws its feed from the top of the accumulator by means of the secondary feed pump controlled either by hand or by the boiler feed water regulator B. The circulating pump draws water either direct from the discharge of the primary feed pump or from the bottom of the accumulator or from both and sprays it into the top of the accumulator. The output of the circulating pump is controlled by the regulator C impulsed by the boiler pressure.

When the boiler pressure rises the regulator C opens, more water is sprayed into the steam space of the accumulator ; this condenses more boiler steam, absorbs the extra output of the boiler and heats the feed water in the accumulator. When the steam demand is low and the regulator is well open the circulating pump will take not only the output of the primary feed pump but will draw cool water from the bottom of the accumulator. If the steam demand drops to such an extent that the primary feed regulator A closes, the circulating pump will draw the whole of its requirements, which will now be a maximum, from the bottom of the accumulator.

If the steam demand rises the accumulator regulator C closes, consequently little cool feed is sprayed into the accumulator and little live steam is condensed for feed heating. When the steam demand is so great that the lowered boiler pressure completely closes the regulator C no water will be circulated, no steam will be condensed in the accumulator, all the steam will be available to meet the demand and the primary feed will go into the bottom of the accumulator pushing the hot water upwards. If the steam demand is extra heavy surface flashing will take place in the accumulator and add to the supply of steam.

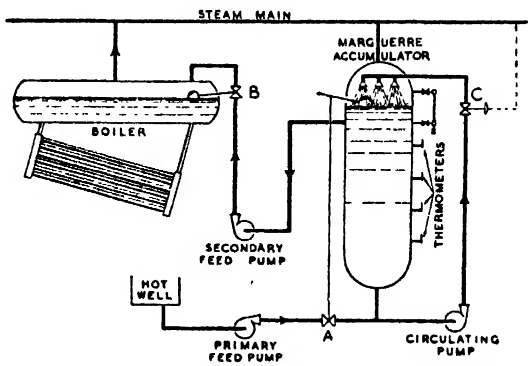

FIG. 265. MARGUERRE FEED WATER ACCUMULATOR

The boiler firing rate is controlled by the temperature level of the hot water in the accumulator.

The accumulator is arranged vertically so that the feed water can remain in layers at different temperatures, the heated feed at the top, from which the secondary feed pump draws its supply ; the unheated feed water in the bottom of the accumulator. One of the advantages of the Marguerre accumulator is that it is much easier to find space in an existing works for a vertical than for a horizontal vessel.

456. LIMITATIONS IN THE APPLICATION OF FEED WATER ACCUMULATORS. Clearly accumulators of this type will be smaller and more "efficient" in a boiler plant where the feed water enters the boilers relatively cool. The arrangement can hardly be called efficient from a thermodynamical point of view, because all the feed heating is done by live steam at boiler pressure. Except in very advanced plants fitted with steaming economisers there is always a part of the feed heating done by means of boiler steam or its equivalent. The less the temperature difference between the feed water and the boiler water the less application is there for a feed water accumulator, or the larger must such an accumulator be, unless the feed is heated by turbine bleeds—this point is dealt with in a minute. Where an economiser brings the feed up to nearly boiler temperature there is little scope for a feed water accumulator. As the feed water accumulator is essentially a plant which is

very suitable for smoothing the peaks out of a power plant and as power plant is generally supplied with well heated feed water, we must see what can be done to enable this excellent plant to be used.

457. FEED WATER ACCUMULATORS AND ECONOMISERS. Fig. 266 shows the method that can be used if the boiler is fitted with economisers. The faster the boiler is steaming the more necessary is it to ensure a good flow through the economiser. The water therefore is pumped by the primary feed pump through the economiser before going either into the top or the bottom of the feed water accumulator. If the economiser puts much heat into the water, this makes the effective storage capacity of the accumulator much less until, with steaming economisers, which can bring the temperature right up to boiler temperature, there is no scope for a Marguerre accumulator at all. In such cases the accumulator must take the Kiesselbach form, but even then it becomes simply a flash vessel which merely increases the water capacity of the boiler. It does not really accumulate heat at slack periods for use during peaks. This does not mean that it does not have good uses. Increasing the water capacity of a boiler is one of the best ways of dealing with peaks especially those that are very short and sharp as has been discussed in Section 435.

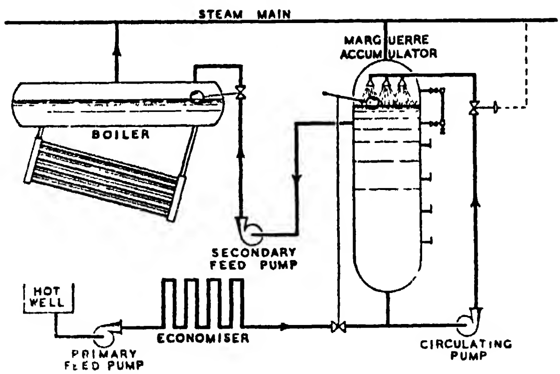

FIG. 266. MARGUERRE ACCUMULATOR WITH ECONOMISER

458. FEED WATER ACCUMULATORS AND BLEED HEATERS. In the case of power plant that is subject to peaks and which has feed water heating by turbine bleeds there is an excellent application for feed water accumulators. Not only is the task of the boiler made more easy but the generating capacity of the turbine can be increased.

Fig. 267 shows the application of a Marguerre accumulator to a system with three regenerative bleeds. It is not necessarily a Marguerre job ; the Kiesselbach type may be equally, possibly more suitable in some cases. The bulk of the feed heating is done by the bleed heaters, only the topping up of temperature being done by the economiser. No live steam is used for feed heating. The steam pressure in the accumulator is that of the first bleed. The regulator C works as in Fig. 265 but it has a double effect. When the load is light a lot of water is circulated through the bleed heaters and the amount of steam bled

will be large. When the load is heavy, the circulation drops, consequently the amount of bled steam falls and there is more steam available to pass right through the turbine and consequently generate more power. At extreme peak loads the circulation stops altogether, and no steam is drawn from the bleeds for feed heating. The system is very beneficial. At light loads when the turbine is normally less efficient than at full load, the cycle efficiency is improved by a greater quantity of steam being bled in order to heat a store of feed water. At heavy loads when the turbine is hard pressed, the bleeds are cut off entirely, and all the steam that would have been bled can pass through the efficient low pressure stages and produce extra power.

If the design of the plant is such that all the flue gas heat can be absorbed by the air heater, then all the feed heating can be done by bleeds, but if the stoker is such that it cannot tolerate a very high air temperature an economiser will be necessary. The economiser should be put between the accumulator and the boiler. The arrangement is shown in Fig. 267.

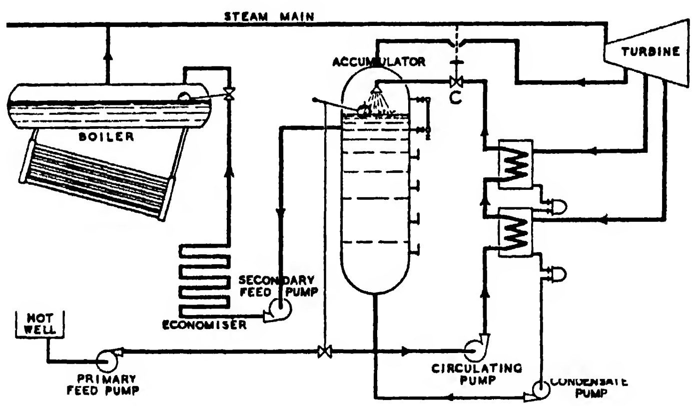

FIG. 267. FEED WATER ACCUMULATOR WITH REGENERATIVE BLEED HEATING

459. PEAK CAPACITY WITH REGENERATIVE FEED WATER ACCUMULATOR. If we turn to Fig. 24 in Section 88, Chapter 2, we see that the boiler feed is heated from the temperature corresponding to 29 in. vacuum to that corresponding to 80 psi by means of three regenerative feed heaters.

The steam table tells us that the heat addition is $294 \cdot 5 - 47 = 247 \cdot 5$ Btu/lb.

This corresponds to area A′Paa′.

If the feed heating bleeds are cut out on peak loads by the accumulator regulator we will secure the additional power area A′PA.

By finding the area A′Aaa′ and deducting it rom $247 \cdot 5$ we can find the gain of power A′PA that will be available on overload.

The entropy at A′, 29 in. vacuum, is ·091.

The entropy at P, 80 psi, is ·469.

The temperature at 29 in. vacuum is 79° F.

Area A′Aaa′ is $(79 + 460) \times (\cdot 469 - \cdot 091) = 203\cdot 7$ Btu,

so that the available power area A′PA is $247\cdot 5 - 203\cdot 7 = 43\cdot 8$ Btu/lb.

Fig. 22 shows the cycle that will be in operation when the bleeds are cut out, so that the extra power of 43·8 will be the difference between the power areas in Figs. 22 and 24.

In Fig. 22 we found that the area ABCC′D was 566 Btu,

so that the power area in Fig. 24 is $566 - 43\cdot 8 = 522\cdot 2$ Btu.

If the overall efficiency of the turbine in Fig. 24 was 82 per cent

the net output of the machine would be 428·2 Btu./lb.

The extra power A′PA is obtained at low pressure and can therefore be assumed to be obtained at an efficiency ratio of 83 per cent., so that the extra power available on overload when the bleeds are cut out will be

83 per cent. of $43\cdot 8 = 36\cdot 4$ Btu.

The effective overload will represent an extra 8·5 per cent.

So that the feed water accumulator relieves the boiler of all feed heating during a peak, and gives the turbine an extra capacity of 8·5 per cent. for the same amount of input steam.

460. EVAPORATOR ACCUMULATOR. In Section 117, Chapter 3, it was pointed out that wherever possible reducing valves should be replaced by plant which gives some return for the loss of virtue that the steam has suffered by its pressure reduction. Three methods were there suggested : engines or turbines ; evaporators ; accumulators. In each of these plants we get a useful return for the increase of entropy that we have permitted to take place. Each of these types of plant have been dealt with separately. It is however possible and sometimes very convenient to combine an accumulator with an evaporator.

In a Ruths accumulator the steam that is blown into the accumulator condenses by contact with the water in the accumulator and gives up its latent heat to the water. It also adds its condensate containing sensible heat to the water volume in the accumulator. The steam could have been put into a heating surface inside the accumulator where it would condense, give up its latent heat to the accumulator water, be discharged from the heating surface through a trap and reappear as hot condensate.

When the accumulator pressure was reduced the water in the accumulator would flash off the latent heat that had been put into it. As no water had been added with this latent heat, the water in the accumulator would gradually evaporate and the quantity would need to be maintained by means of some feed arrangement.

Many process factories collect their process condensate for boiler feed. There is always some loss due to leakage from safety valves and due to any steam that is directly injected into the process. Sometimes some of the process condensate is contaminated with the process material and must be discarded. It may, therefore, be very beneficial to make distilled water to provide the necessary make-up. This is what the evaporator accumulator does.

There are objections to putting the heating surface inside the accumulator body. If the water that is to be distilled is hard there will be scale formation sooner or later, dependent on the efficacy of the water treatment. It is difficult to arrange, inside the accumulator body of orthodox shape, a good heating surface which will promote the very high rate of circulation that is wanted.

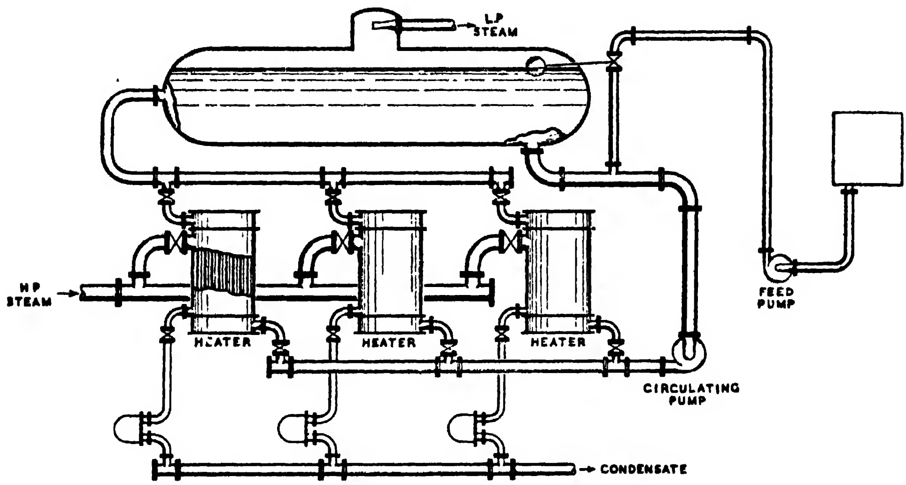

FIG. 268. EVAPORATOR ACCUMULATOR

Fig. 268 shows a suitable arrangement. The accumulator water is circulated, by means of a pump, through the three heaters or condensers shown. As the water takes up heat its pressure rises, the charging steam condenses and the condensate is removed by the traps. When a lowered pressure on the accumulator causes flash, the necessary steam is discharged and the water in the accumulator is maintained at the correct level by the water level regulator and the feed pump. If scaling occurs in the heaters, one heater at a time can be taken out of service and descaled without putting the plant out of action. The control gear can be exactly the same as on a Ruths accumulator.

461. THE STORAGE OF STEAM AS STEAM. We have seen in Sections 447 and 450 that an exhaust accumulator which stores the heat in water must operate over a pressure drop of something within the practical range of 1·5 psi to 6 psi. There may quite often be circumstances where a pressure drop is most undesirable or at least must be reduced to the lowest possible figure. To meet such conditions it is possible to store steam as steam at constant pressure.

462. GAS-HOLDER TYPE ACCUMULATORS. Accumulators of the gas-holder type have been used to a small extent, more especially in Continental Europe. There is much to be said against the gas-holder design, but there is also quite a lot to be said in its favour. It is large, it is expensive, the maintenance is appreciable, the heat loss is considerable, but it is simple,

there is no mystery about it, by looking at it we can see whether it is working correctly or not, and—it stores steam with at most ½ psi pressure drop which may be of paramount importance.

Let us take the winding engine exhaust considered in Section 448 and see what kind of a gas-holder will be necessary. The excess of exhaust over usage is 430 lb. of steam per wind. The holder must therefore have a capacity of about 450 lb. The pressure required to lift the first lift of a gas holder is about ·2 psi rising to about ·5 for the third lift. The holder for handling the winder peak will be quite small and the holder can probably be made with a single lift, which, owing to its small size, may need a pressure of ·5 psi to raise it.

At ·5 psi.g. the volume of 1 lb. of steam is 26 cu. ft. so that the holder must have a volume of $450 \times 26 = 11{,}700$ cu. ft. A holder 26 feet in diameter and 24 feet high will have a capacity of 12,740 cu. ft. which gives a little margin.

The area of the roof will be 531 sq. ft. If we say that on average the holder will be 15 ft. extended,

the area of the exposed side will be $82 \times 15 = 1{,}230$ sq. ft.

As the bottom will lose heat we can call the radiating surface 1,850 sq. ft.

Table XXII tells us that the heat loss at 212° F. will be 330 Btu/sq. ft./hr. in still air at 70° F. This is a temperature difference of 142° F.

We can assume that the heat loss at 45° F. will be $\dfrac{330 \times 167}{142} = 388.$

The amount of steam condensed will be $\dfrac{388 \times 1{,}850}{971} = 740$ lb./hr.

The amount of steam that passes through the accumulator per hour is $430 \times 40 = 17{,}200$ lb./hr. The heat loss represents 4·3 per cent.

Suppose we have an outside temperature of 20° F. and a 20 m.p.h. wind (see Table XXVI) the steam condensed will be

$$\frac{330 \times 192 \times 1{,}850 \times 4}{142 \times 971} = 3{,}400 \text{ lb./hr. or } 19{\cdot}8 \text{ per cent.}$$

for very exceptional conditions. These are very reasonable losses, and we do at least recover the distilled water from the steam that has lost its heat. Oil in the exhaust is actually beneficial, not harmful. After a short time in action accumulators of this type quickly get a most effective insulating coating of thick oil.

As regards controls, there are certain peculiar qualities possessed by this accumulator which call for special measures. In a water accumulator the discharge rate gradually diminishes and finally fades out almost imperceptibly, and during discharge the pressure is slowly falling, thus delivering a message to the mixed pressure turbine that the supply is running out.

In the case of the steam holder the pressure is the same whether the holder is full or almost empty and when it is empty the supply just stops. In such an event there would be a moment's delay in the high pressure turbine governor coming into action and the machine might drop more speed than would be desirable.

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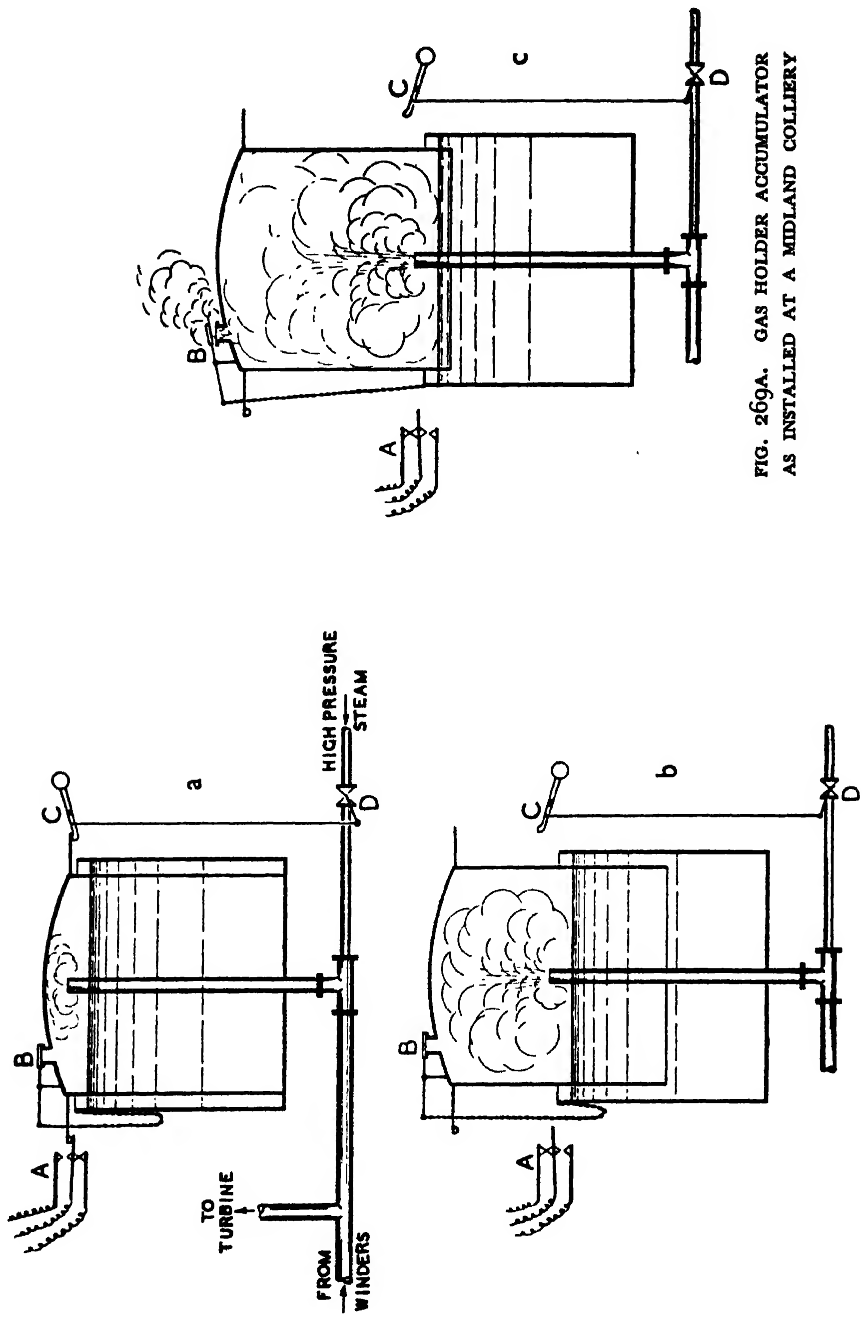

FIG. 269A. GAS HOLDER ACCUMULATOR AS INSTALLED AT A MIDLAND COLLIERY

An installation at a British colliery will be described, and an alternative control will then be suggested. The pit is shallow and the winds are consequently short. The accumulator takes the exhaust from two winding engines and feeds a mixed pressure turbine. The turbine requires considerably more steam than is provided by the exhaust from the winders. The controls work thus : When the accumulator starts to rise a switch A in Fig. 269A is moved and puts the high pressure turbine governor out of action and puts the low pressure governor into operation. This is the position shown in Fig. 269Ab. The accumulator continues to rise and, normally, before it is in its fully extended position the winder has stopped. If, however, both winders happen to run exactly together there might be a risk of the holder playing the part of secret weapon. To guard against this the weighted valve B is pulled open by a chain which tightens when the holder approaches the danger height, thus releasing the steam—see Fig. 269Ac. When the winders stop, the holder empties rapidly and means must be provided for preventing the lift dropping into the tank with a bang. This is done by the lever C being operated by the descending holder. Lever C opens a small valve which admits high pressure steam to the holder and cushions the lift. This is shown in Fig. 269Aa. When the lift has dropped almost home the switch A changes over from the low pressure governor to the high pressure governor.

Now this holder has been going up and down about 40 times an hour for 31 years. If only one winder comes on at a time it only goes part way up. Very early in its history valve D went wrong, and that part of the control has never worked satisfactorily. It seems also that switch A could be dispensed with if the turbine could be told in some other way that the supply was running short.

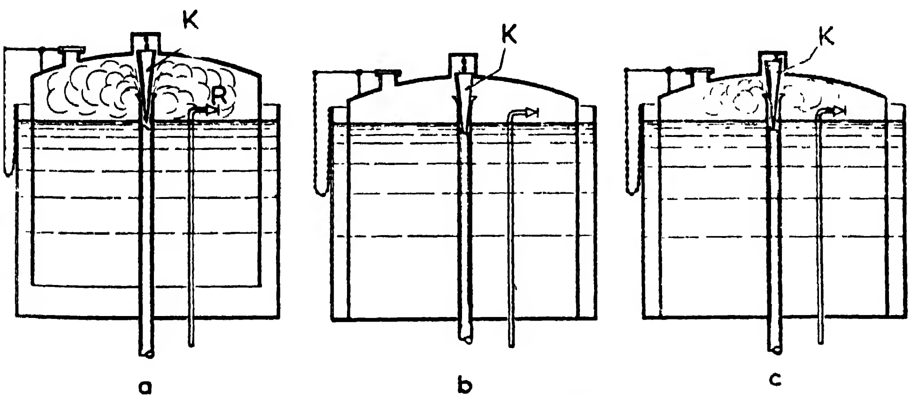

FIG. 269B. SUGGESTED ARRANGEMENT OF GAS HOLDER ACCUMULATOR

The arrangement shown in Fig. 269B might solve both these problems in the simplest possible way. The end of the steam pipe is flared and a long gently tapering " cork " K which can enter the steam pipe is suspended by a short chain from the top of the lift. The holder in Fig. 269Ba is shown discharging. The cork is just entering the steam pipe and thus reducing the steam discharge

rate. This throttling of the steam gives a message to the turbine whose low pressure governor will try to get more steam. In this it will fail and will in fact get less, so that the high pressure governor must open. As the lift falls the discharge rate is progressively reduced, thus slowing down the fall of the lift until, in the position shown at Fig. 269Bb, the outlet is plugged and the lift is just clear of the bottom.

When the winders start the cork K will be blown out as in Fig. 269BC and steam can enter the holder freely.

Should there be a hitch in winding, the steam inside the holder in the down position would condense and create a vacuum inside. To protect the holder against collapse, a reducing valve R from a small high pressure steam pipe can be arranged to open at about 2 in. or 3 in. vacuum.

The cost of a steam holder will be greater than that of the exhaust Rateau accumulator discussed in Section 448. The difference in cost together with the extra maintenance must be set against the fact that with a back pressure of ·5 psi instead of about 2·5 psi the winding engine will use probably 4 per cent. less steam, or the performance of the exhaust turbine will be correspondingly improved by loading the steam holder to a 2 psi pressure.

463. GAS-BAG ACCUMULATORS. There would seem to be no good reason why steam should not be stored at atmospheric pressure in a fine fabric bag. The back pressure would only be a few inches of water. The heat loss would be small as the bag could be encased in a cheap insulating box. Very thin cotton fabrics can now be woven that are almost impervious when wetted. The cost would be low.

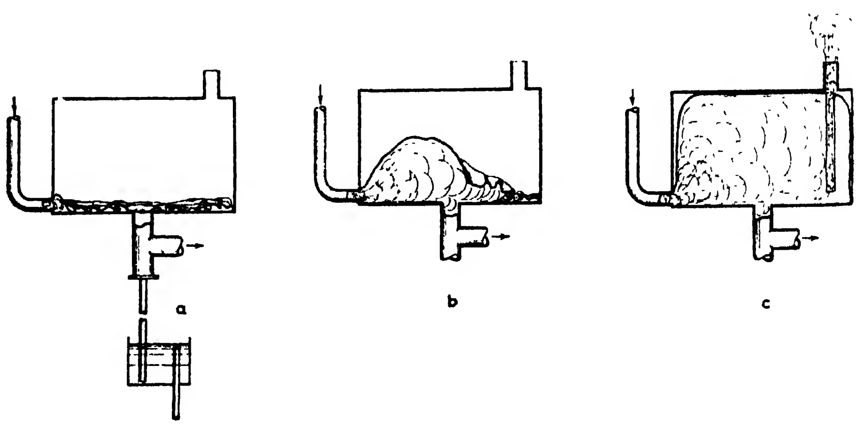

FIG. 270. GAS BAG ACCUMULATOR

Fig. 270 shows the suggestion. The bag would be roughly rectangular. When fully distended in its box, one corner would raise a collapsible cotton tube distended with light hoops which would then release any surplus steam.

As the corner to which the release tube was fastened would be the heaviest part of the bag it would be the last to lift. As far as the author knows, such a bag has never been tried, nor can he hazard a guess as to how long it would last. Such a device would need controls similar to those on the gas holder type.

Where steam is stored as steam and passes to an exhaust or mixed pressure turbine, there should be an oil separator in the pipe leading to the turbine to minimize deposition of oil on the blades, particularly if these are of the reaction type.

464. HOT PROCESS WATER STORAGE. If we store our steam heat in the water in a Ruths accumulator or a Rateau accumulator we recover the heat as low pressure steam. If we store our steam heat in the water of a Kiesselbach or Marguerre accumulator we recover the heat as high pressure steam in increased boiler output at peaks. If we store our heat in hot process water or hot heating water we recover the heat as extra boiler output when the demand is heavy.

Many factories require large amounts of hot water which must often be heated up from cold. Cold water requires a great deal of heat and, pound for pound, can absorb much more heat than the hot water in an ordinary accumulator which operates over a small temperature difference.

We see from Table IV that it requires 100 lb. of water at 212° F. to store 1 lb. of steam at 3 psi.

Now the amount of 3 psi steam that 100 lb. of water at 60° F. will absorb when heated to 210° F. is found if we put $x =$ to the weight of 3 psi steam and equate thus :

$$100 \times (60 - 32) + 1{,}154x = (x + 100)(210 - 32)$$
$$x = 15{\cdot}4$$

So that, when water must be heated from cold, a water tank need only be one-fifteenth the size of an equivalent orthodox exhaust accumulator.

In laundries, breweries, etc., great quantities of hot water are required. In a brewery, for example, the mashing liquor calls for much heat for a few hours at the beginning of the brew. The vapour from the copper contains quite enough heat to heat all the mashing liquor, but this source of heat is only available 3 or 4 hours after it is wanted. If the mashing liquor is heated by copper vapour the hot liquor must be stored for perhaps 20 hours. In many other factories the boilers could very conveniently heat water in their off-load periods, if the hot water could be economically stored.

The storage of heat in hot process water is one of the best ways of smoothing long-term peaks and one of the best ways of recovering waste heat, but its whole success or failure depends on how much heat will be lost by the water tank during long periods of storage.

It is desirable therefore to have an easy method of calculating how much heat will be lost by the contents of a tank under various conditions. The problem is not straightforward like the calculation of heat loss from a tank that is

kept at a constant temperature. The temperature of a storage tank such as we are considering is falling at a gradually diminishing rate and the equation for the process is :—

$$d = \theta_1 (1 - e^{-tk})$$

where d = loss of temperature of tank contents in ° F.

t = timein hours the tank is cooling.

$$k = \frac{UA}{WS}$$

W = Weight in lb. of tank contents.

S = Specific heat of tank contents.

U = Heat loss from tank surface in Btu/sq. ft./°F./hour.

A = Area of tank surface in square feet.

θ_1 = Initial temperature difference between tank and air in °F.

This equation is exact but is in an inconvenient form.

The author's brother has devised a simple empirical formula which is accurate within 1 per cent. over the range of conditions likely to be met in practice.

This empirical equation is

$$d = \theta_1 \left(\frac{1 \cdot 65tk}{1 \cdot 6 + tk}\right)$$

The error introduced by using this empirical formula instead of the accurate formula never exceeds 1 per cent. provided the value of tk is less than 1·0.

Suppose we have a rectangular tank 6 ft. × 12 ft. × 4 ft.

Its capacity will be 288 cu. ft. exactly
or say 1,750 gallons for practical purposes.

The surface of its six sides is 288 sq. ft. (It is pure coincidence that this figure is the same as the volume).

Assume that the tank is covered and lagged all over.

Assume that the tank contains water.

Assume that the initial water temperature is 210° F. or 180° F. or 150° F.

Assume that the air temperature is 70° F.

Assume that the tank cools for 6, 12, 18 and 24 hours.

Assume that the tank contains 500 or 1,000 or 1,500 gallons.

Then t = 6, 12, 18, 24.

W = 5,000, 10,000, 15,000.

S = 1·0.

U = ·5.

A = 288.

θ_1 = 140, 110, 80.

We will first calculate the values of tk for the various conditions to see that they do not exceed 1·0 and we can put these down in Table A.

In no case does tk exceed 1·0 so that the empirical formula is applicable over the whole range in question. We can now work out the loss of temperature of the water under various conditions of initial temperature, see Table B.

TABLE A

		tk			
W	k \ t	6	12	18	24
5,000	·0288	·1728	·3456	·5184	·6912
10,000	·0144	·0864	·1728	·2592	·3456
15,000	·0096	·0576	·1152	·1728	·2304

TABLE B. TEMPERATURE LOSS OF WATER IN A RECTANGULAR TANK 6 FT. × 12 FT. × 4 FT., COVERED AND LAGGED, IN °F. WITH AIR AT 70° F.

TANK CONTENTS lb.	θ_i = 210° F.				θ_i = 180° F.				θ_i = 150° F.			
	6 HR.	12 HR.	18 HR.	24 HR.	6 HR.	12 HR.	18 HR.	24 HR.	6 HR.	12 HR.	18 HR.	24 HR.
5,000	22·5	41·0	56·5	69·7	17·7	32·2	44·4	58·8	12·9	23·4	32·3	39·8
10,000	11·8	22·5	32·2	41·0	9·3	17·7	25·3	32·2	6·8	12·9	18·4	23·4
15,000	8·0	15·5	22·5	29·1	6·3	12·2	17·7	22·8	4·6	8 9	12·9	16·6

Now the tank we have been considering is not a good shape. Its surface is large compared to its content. Let us take a more economical shape, namely a cylindrical tank 7 ft. diameter, 7 ft. 3 in. high. This will have the same volume as the 6 ft. × 12 ft. × 4 ft. rectangular tank, but its heat losing surface will be only 236·5 sq. ft.—see Table C. (The tank shape having the lowest surface/volume ratio has a height = diameter).

TABLE C. TEMPERATURE LOSS OF WATER IN A CYLINDRICAL TANK 7 FT. DIA. × 7 FT. 3 IN., COVERED AND LAGGED, IN ° F. WITH AIR AT 70° F.

TANK CONTENTS lb.	θ_i = 210° F.				θ_i = 180° F.				θ_i = 150° F.			
	6 HR.	12 HR.	18 HR.	24 HR.	6 HR.	12 HR.	18 HR.	24 HR.	6 HR.	12 HR.	18 HR.	24 HR.
5,000	18·8	34·9	48·7	60·7	14·8	27·4	38·3	47·7	10·8	19·9	27·8	34·7
10,000	9·8	18·8	27·0	34·9	7·7	14·8	21·2	27·4	5·6	10·8	15·4	19·9
15,000	6·7	12·9	18·8	24·5	5·2	10·2	14·8	19·2	3·8	7·4	10·8	14·0

This cylindrical tank shows a very definite improvement on the rectangular tank. But such a tank is of little use for any moderate sized laundry, dye works or brewery. Let us take a tank that will hold 10,000 gallons, namely one 13 ft. diameter, 12 ft. 6 in. high.

The volume is	1,659 cu. ft. or 10,339 gallons.
The wall area is	510·51 sq. ft.
Top and bottom	132·73 sq. ft. each.
Total surface	776 sq. ft.

We will assume the top is covered and the whole tank is lagged—see Table D.

TABLE D. TEMPERATURE LOSS OF WATER IN A CYLINDRICAL TANK 13 FT. DIA. × 12 FT. 6 IN., COVERED AND LAGGED, IN ° F. AIR AT 70° F.

TANK CONTENTS lb.	$\theta_i = 210°$ F.				$\theta_i = 180°$ F.				$\theta_i = 150°$ F.			
	6 HR.	12 HR.	18 HR.	24 HR.	6 HR.	12 HR.	18 HR.	24 HR.	6 HR.	12 HR.	18 HR.	24 HR.
25,000	12·7	24·1	34·3	43·6	10·0	18·9	27·0	34·3	7·3	13·7	19·6	24·9
50,000	6·5	12·7	18·5	24·1	5·1	10·0	14·6	18·9	3·7	7·3	10·6	13·7
75,000	4·4	8·6	12·7	16·6	3·5	6·8	10·0	13·1	2·5	4·9	7·3	9·5
100,000	3·3	6·5	9·7	12·7	2·6	5·1	7·6	10·0	1·9	3·7	5·5	7·3

Table D shows the great advantage of a large tank over a small one, Table C. But it should be noted what a big temperature drop there is when the tank is only partly full.

Now many, perhaps most, tanks are not lagged on their bottoms which generally rest on joists. If the tank is sitting on a good thick floor its bottom is virtually lagged, but on joists it can lose a lot of heat.

Too many tanks are uncovered. The heat loss from a hot liquid surface is much greater than the heat loss from a similar area of metal tank. This is because quite a lot of evaporation takes place at the surface and this absorbs much latent heat. There is very little available information on the heat lost from the liquid surface in an open tank. There are wide variations. If the liquid surface is in a draught the loss is greatly increased. If the tank is full, the heat loss from the liquid surface is much greater than if the tank is only part full. When the level is low there is a deep blanket of saturated air over the surface which limits the rate of evaporation.

Let us try to compare the heat that may be lost from the open top of the 13 ft. tank with that which might be lost from its unlagged side.

One of the simplest published formulae for heat loss from water surfaces is that given by Carrier for cooling ponds and which appears in some of the books.

$$Q = \cdot 093\, L \left(1 + \frac{V}{230}\right) (p_s - p)$$

where Q = Heat lost from the surface in Btu/sq. ft./hr.
L = Latent heat at the water temperature.
V = Air speed across the surface in ft./min.
p_s = Saturation pressure of steam at the water temperature in inches Hg.
p = Partial pressure of the water vapour in the air reaching the surface in inches Hg.

Now p depends on the humidity and temperature of the air and is usually variable and generally unknown. It may lie between ·3 in. and ·7 in. depending on the conditions. It is probably safe enough to call it ·4 for average factory conditions.

V is also extremely variable, is unpredictable and exceedingly difficult to measure accurately. For simplicity we will take it at 230 ft./min. or about 2½ m.p.h. which is probably not far out for average conditions.

The water surface of the 13 ft. tank is 132·7 sq. ft.

The cylindrical tank side area is 510·5 sq. ft.

We can now work out the heat loss from the water surface and the loss from the unlagged side (at 2 Btu/sq. ft./° F./hr.) and see their relative importance.

TABLE E. HEAT LOSS IN ONE HOUR FROM WATER SURFACE AND FROM UNLAGGED SIDE OF TANK 13 FT. DIA. × 12 FT. 6 IN. WITH AIR AT 70° F.

Average tank temperature ° F.	200	180	160	140
L = latent heat	978	991	1,002	1,014
p_s = saturation pressure inch Hg.	23·5	15·3	9·7	5·9
$p_s - p = p_s - \cdot 4$	23·1	14·9	9·3	5·5
Q = Btu/sq. ft./hour	4,202	2,746	1,733	1,037
Q × 132·7 = water surface loss	557,600	364,450	230,000	137,650
Average temperature difference ° F.	130	110	90	70
Temperature difference × 2 × side area = side loss	132,730	112,310	91,890	71,470
Surface loss/side loss	4·2	3·2	2·5	1·9

Table E shows that it is about 2 to 4 times as important to close the top of the tank as to lag the sides. The figures in Table E must not be looked on as exact. They are not ; they are simply indicative of the order of magnitude of the loss from water surfaces. Measurements in the author's factory show that they are not far out.

One of the troubles with covering hot tanks is the corrosion that occurs in steel. This corrosion is one of the principal reasons why more tanks are not covered. But tanks should be covered and the corrosion should be avoided by using stainless steel, monel metal or simply brass, copper or wood.

Over short periods there is no need to adopt the formula used here for estimating the tank heat loss. The temperature drop is small and the temperature difference can be assumed to be constant over the period without introducing much of an error. But over long periods such a method of estimating temperature drop introduces a big error, in the one direction of showing the temperature drop to be larger than it actually would be. This may dissuade the management from installing hot water storage which might really be of great benefit.

Table F gives a comparison of the two methods of estimating temperature loss in the 13 ft. tank containing 10,000 gallons at an initial temperature of 210° F., with the tank closed and lagged and with the tank closed but unlagged.

TABLE F

TIME OF COOLING	TEMPERATURE DROP OF WATER FROM 210° F. WITH AIR AT 70° F.			
	DIMINISHING TEMPERATURE DIFFERENCE $\theta_i\left(\frac{1{\cdot}65tk}{1{\cdot}6+tk}\right)$		CONSTANT TEMPERATURE DIFFERENCE $\theta_i\, tk$	
	LAGGED	UNLAGGED	LAGGED	UNLAGGED
6	3·3	12·7	3·3	13·0
12	6·5	24·1	6·5	26·1
18	9·7	34·3	9·8	39·1
24	12·7	43·6	13·0	52·1
30	15·7	52·1	16·3	65·1
36	18·5	59·8	19·6	78·1
42	21·4	66·9	22·8	91·2
48	24·1	73·4	26·1	104·3

The necessity of using the diminishing temperature difference is clearly shown in Table F if the heat losses are great, or the tank small. By taking a constant temperature difference it appears that nearly three quarters of the heat in the water in the unlagged tank would be lost. Taking the diminishing temperature difference we see that in actual fact only about half the heat will be lost and week-end hot water storage, even in the unlagged tank, might pay a dividend.

It will also be seen that where the heat loss is small and the tank large, the error introduced by using the constant temperature difference is very small, smaller probably than the error in estimating what heat loss/sq. ft./° F./hr. to use as a basis.

This section is already very long, but no excuse is offered for making it still longer, because the storage of heat as hot water is so cheap, so simple, so universally applicable, that it deserves the highest priority in thermal projects.

One last example will now be considered to hammer home the claims of hot water storage to be one of the main weapons in the thermal technologists' armoury.

We will take the same 13 ft. tank as before, but in addition to covering its top we will lag it more elaborately. In Section 138 a long steam pipe connecting two Liverpool factories was shown to have a heat loss of only ·3 Btu/sq. ft./° F./hr., while the steam accumulator in the author's factory was shown, in Section 446, to have a heat loss of only ·175 Btu/sq. ft./° F./hr. We will assume that the tank is so well lagged that it has a heat loss of ·25 Btu/sq. ft./° F./hr.

TABLE LXIA. TEMPERATURE LOSS IN °F. FROM 10,000 GALLONS WATER IN CYLINDRICAL TANK 13 ft. DIA. & 12 ft. 6 in. HIGH, COVERED AND HEAVILY LAGGED.

TIME OF COOLING		TEMPERATURE DROP OF WATER FROM 210° F.	
HOURS	DAYS	AIR AT 70° F.	AIR AT 50° F.
		° F.	° F.
6		1·7	1·9
12		3·3	3·8
18		4·9	5·6
24	1	6·5	7·5
30		8·1	9·3
36		9·7	11·0
42		11·2	12·8
48	2	12·7	14·5
54		14·2	16·2
60		15·7	17·7
66		17·1	19·6
72	3	18·5	21·7
78		19·9	22·8
84		21·3	24·4
90		22·7	26·0
96	4	24·1	27·5
102		25·4	29·0
108		26·7	30·3
114		28·0	32·0
120	5	29·3	33·5
126		30·6	35·0
132		31·9	36·5
138		33·1	37·8
144	6	34·3	39·3
150		35·5	40·6
156		36·7	42·0
162		37·9	43·3
168	7	39·1	44·6
174		40·2	45·9
180		41·4	47·3
186		42·5	48·6
192	8	43·6	49·9

(87664) P* 2

Table LXIA shows how remarkably small would be the heat loss even over many days. It is clear that a well lagged water storage tank might be a godsend to those factories who waste a lot of steam at the shut-down on Saturday. This steam could heat cold water for use on Monday.

Fig. 271 shows in striking pictorial form the heat loss incurred every hour from the tank just considered, when open and unlagged, and when covered and lagged.

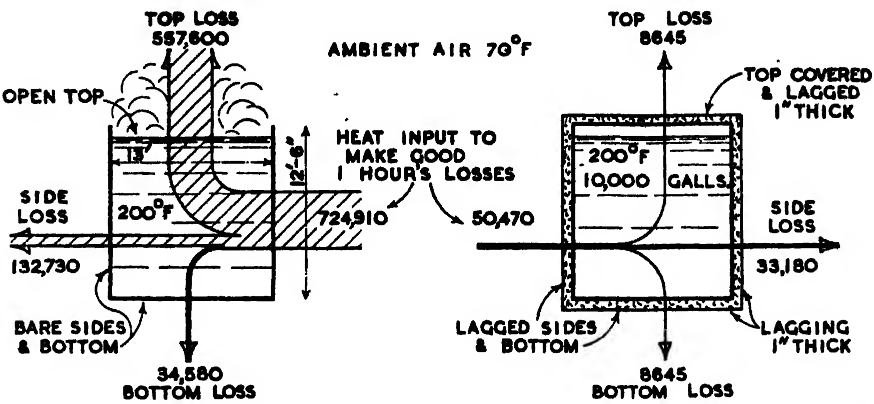

FIG. 271. HEAT LOSSES FROM ROUND OPEN UNLAGGED TANK AND FROM COVERED LAGGED TANK

The lessons to be learnt from the foregoing are :—

1. That a hot water storage tank should always be so sized that it is as nearly full as possible.
2. That it is of the greatest importance to cover the top of the tank.
3. That a storage tank should be very well lagged.
4. The larger the tank the more is elaborate lagging justified.

465. ACCUMULATOR APPLICATIONS. In any one factory having peak demands or peak exhausts there are all kinds of ways in which accumulators can be applied. Often it is by no means obvious where the accumulator should be inserted in the pressure hierarchy, consequently it is not certain which type of accumulator should be used.

In the following sections a few examples are given of the ways in which the various types of accumulator can be applied and a few remarks are added to show the reasons for the choice. These are examples pure and simple, they are not dogmatic proposals for particular industries. Every single plant has its own peculiar problems and local circumstances, and each plant must be considered in the light of its own particular needs.

466. COLLIERY ACCUMULATOR. In a colliery, apart from the winding engine which is one of the peakiest kinds of load, there is a considerable steady load in ventilation and pumping. The winding engines call on the boilers for

sudden huge supplies of steam with the least possible pressure drop. The engine produces sudden huge peaks of exhaust steam. This exhaust steam, could it be stored, would provide much of the steady load if fed to an exhaust turbine.

Fig. 272 shows a small installation where Lancashire boilers, with their large water capacity, are able to make a fair showing at meeting the winder demands. The exhaust from the winding engine goes to a Rateau accumulator and thence into an exhaust condensing turbine. A reducing valve E keeps the turbine going in the event of a temporary hitch in winding.

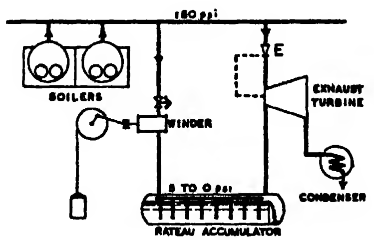

FIG. 272. RATEAU ACCUMULATOR IN SMALL COLLIERY

Fig. 273 shows a larger, more modern colliery with higher pressure water-tube boilers. Such boilers have a small water storage and are not so well placed as shell boilers to meet the winding peaks. The boiler plant therefore is fitted with a Kiesselbach accumulator, which, when the peak comes, provides hot feed water at boiler temperature and provides flash steam at boiler pressure. The exhaust from the winders goes into a Rateau accumulator which feeds a mixed pressure turbine.

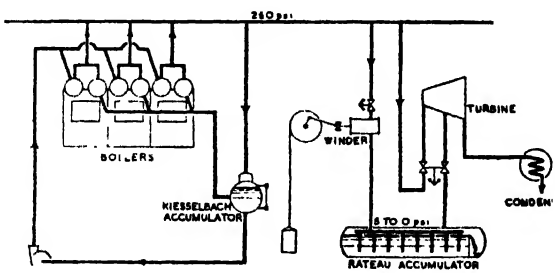

FIG. 273. KIESSELBACH AND RATEAU ACCUMULATORS IN LARGE COLLIERY

467. SUGAR REFINERY ACCUMULATORS. Fig. 274 shows a small sugar refinery where the technique is a little old-fashioned, consequently a relatively large amount of steam is used. This enables the engines taking steam

at 150 psi and exhausting to process at 25 psi to generate sufficient power. The process normally takes rather more steam than is exhausted from the engine so that in this case there is no doubt as to the best place for an accumulator, and this place fixes the type. It will be a Ruths accumulator between the 150 and 25 psi mains. As the peaks are not very severe it is not necessary to use the whole pressure drop in the accumulator which will give enough storage between 80 and 25 psi. Steam for the 5 psi main is provided by a reducing valve from the 25 psi main.

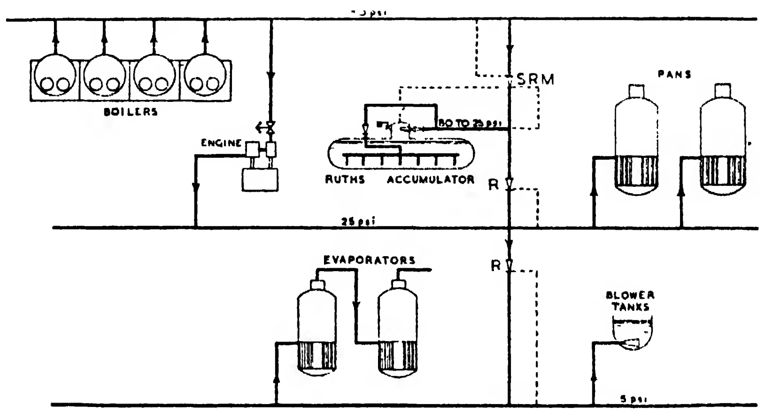

FIG. 274. RUTHS ACCUMULATOR IN SMALL SUGAR REFINERY

Fig. 275 shows a more modern sugar refinery where the white sugar pans take steam at 60 psi. In consequence of this higher back pressure and because of a good technique the process can barely consume the turbine exhaust. The reducing valve E is for emergencies only. In order to generate the needed power the boilers must feed the turbine at 500 psi. The peak demands are produced by the 60 psi and by the 10 psi users and are fairly long-term peaks. They are too long to be met by any simple form of water flash and a big pressure drop is not desirable as it would aggravate the power/steam ratio. A Marguerre accumulator is therefore fitted to the boilers.

Now at times of process valleys the process cannot absorb the turbine exhaust so that the 60 psi safety valve would blow. Clearly the place for an accumulator is on the 60 psi main to absorb the excess turbine exhaust and store it against a future process demand. Although this is an exhaust accumulator it is not feeding an exhaust turbine, but a process main at 10 psi. It therefore need not be large and of Rateau design but can be a Ruths type working over the full pressure drop 60 to 10 psi.

In a sugar refinery there are occasions when the process condensate is a little sweet. Sweet water is not permissible in a boiler working at 500 psi, and such water must be diverted to process. Some of the steam is used direct in blowers. There is therefore always a shortage of condensate for boiler feed

and a pure make-up is very necessary. So here is an ideal application for the evaporator accumulator, which provides pure distilled make-up water, accepts and stores surplus turbine exhaust, takes care of the peaks on the 10 psi main and the valleys on the 60 psi main.

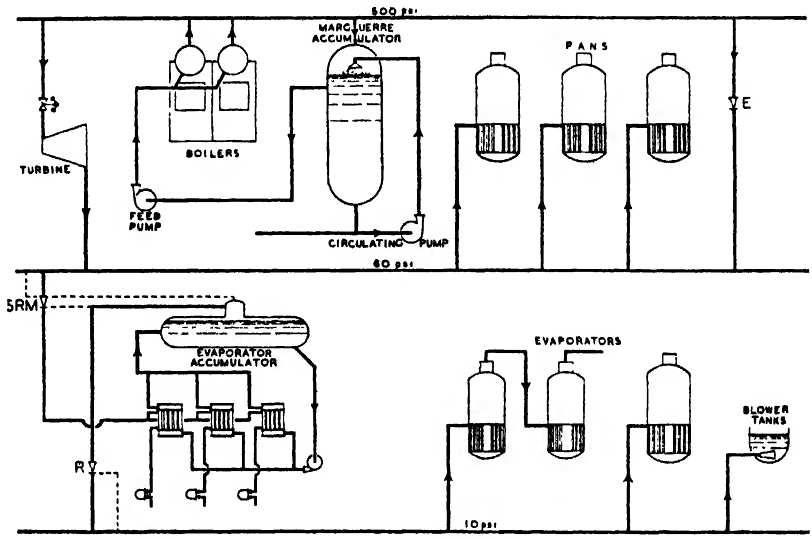

FIG. 275. MARGUERRE AND EVAPORATOR ACCUMULATORS IN LARGE SUGAR REFINERY

468. STEEL WORKS ACCUMULATOR. The rolling mills and hammers demand very sudden, very sharp steam supplies. Water capacity in the boilers is the best way of meeting such a demand. As the pressure is fairly high and the installation is large, the boilers are water-tube and have little inherent water storage. So a Kiesselbach accumulator is the best choice for trying to meet the peaks—see Fig. 276. The steam users are scattered fairly far apart, the exhaust pipes are long so that back pressure makes itself felt and the extra 3 or 4 psi needed by a Rateau accumulator would be undesirable, so a gas-holder type

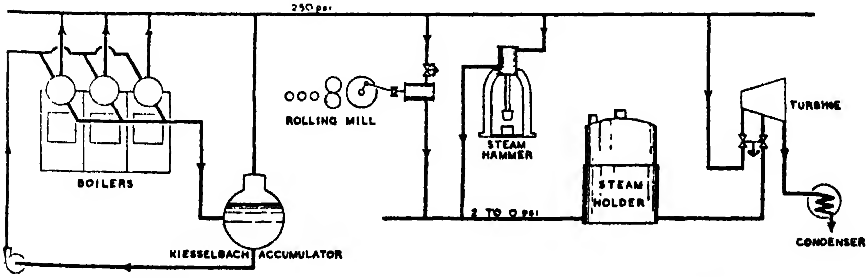

FIG. 276. KIESSELBACH AND GAS HOLDER ACCUMULATORS IN STEEL WORKS

of accumulator is used to collect and even out the supply to the mixed pressure turbine. The high pressure supply to this turbine is only for use in the emergency when chance ordains that all the engines and hammers occupy the same valley.

469. VALUE OF ACCUMULATORS. Accumulators whether high pressure Kiesselbach or Marguerre, medium pressure Ruths, low pressure Rateau, or simply the humble hot water tank are great steam savers. It is often difficult to be sure that the somewhat heavy cost of an accumulator is justified. The

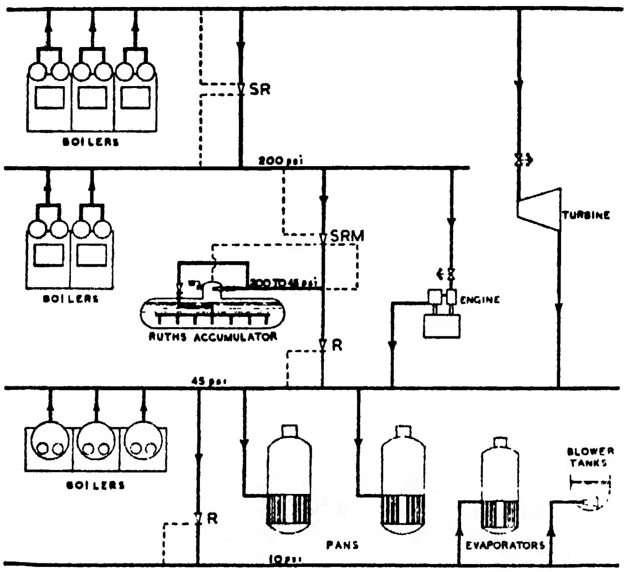

FIG. 277. RUTHS ACCUMULATOR IN OLD SUGAR REFINERY

author is of the opinion that in cases of doubt the verdict should be in favour of installing the accumulator. While it may be difficult to justify an estimated smoothing of peaks, experience seems to be that, once installed, a factory would never part with its accumulator. In this connection the experience in the author's factory may be of interest.

Expansion in a very cramped site had reached the point where two tiers of boilers one above the other were unable to supply the demand. No one was bold enough to suggest a third tier of boilers, but it was thought that if the boilers could be kept on full load continuously the demand could be met. The accumulator was installed as shown in Fig. 277. Some years later progress demanded the complete renewal of the boiler and power plant. It had been assumed that there would be no place for the accumulator in the new scheme, but very few months working of the new plant proved that the accumulator could serve a very useful purpose. It was brought out of retirement and justified itself at once under its new conditions. All the process staff look upon the

" Bag ", as it is affectionately called, as indispensable. The new arrangement is shown in Fig. 278. A recent chart taken of the accumulator and process pressures is shown in Fig. 279.

As a matter of technical interest the story of the putting to work of this accumulator is interesting, illuminating and depressing. At the time of its installation in 1932 it was the largest single accumulator vessel that had been built. It is 14 ft. 9 in. in diameter by 82 ft. long and it is made of plates 1⅛ in. thick.

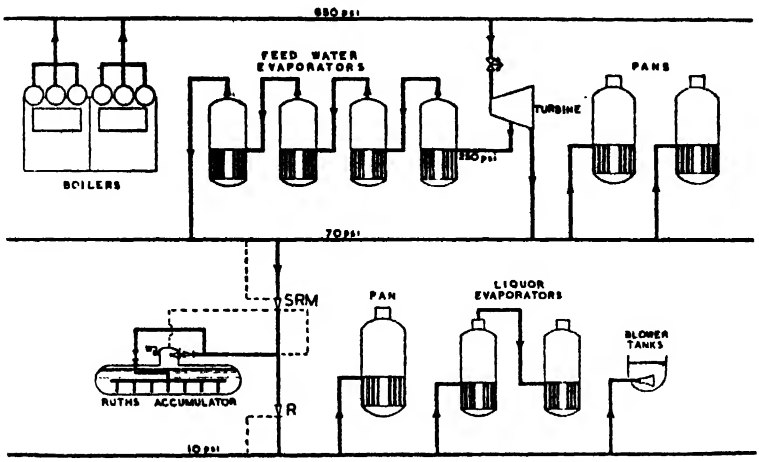

FIG. 278. RUTHS ACCUMULATOR IN MODERNISED SUGAR REFINERY

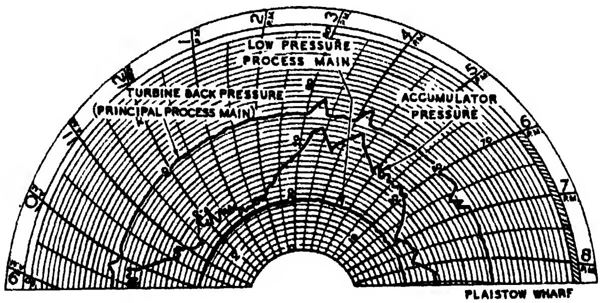

FIG. 279. PRESSURE CHART OF ACCUMULATOR AND PROCESS PRESSURES

Owing to its size it had to be built on site. This called for the closing of many hundreds of 1 $\frac{5}{16}$-in. rivets by hand. This difficult job was done and the joints were well and truly caulked inside. The Insurance Company then called for an hydraulic test at 425 psi. Just before test pressure was reached there was a loud bang—the beautiful caulked joints had sprung an invisible but leakable amount. It was tested cold at 425 psi when it was going to work hot at 250 psi. The test had completely spoiled this beautiful piece of craftsmanship, which leaked ever after until, when it was brought back into low pressure service, all the seams were lightly welded. It cost £13,000 and it saved £5,000 a year.

CHAPTER 17

MULTIPLE EFFECT EVAPORATION

Find out the cause of this effect.
SHAKESPEARE. *Hamlet.* 1601.

THE Multiple Effect Evaporator was invented by Rillieux in 1843 because the exhausted cane fibre provided insufficient fuel in cane sugar factories, and wood had to be cut and carried from increasingly long distances to make up the balance. Nowadays, such is the efficiency attained by the application of Rillieux' invention, that a modern cane factory can not only raise all the steam it needs and generate all the power it requires by burning its exhausted fibre, but has a considerable surplus that makes excellent wall-board.

The Multiple Effect Principle consists of the re-use of latent heat for successive evaporations and is one of the greatest single aids to steam economy, ranking with Watt's invention of the separate condenser. Its application is limited, but it deserves close study because, even if it is impossible to use it, the analysis of its working gives such perfect examples of correct and incorrect steam technique. Once the fundamentals of the technique are mastered there are dozens of ways in which they can be applied in all kinds of directions and which give simple, large and lasting economies.

470. THE SPLITTING-UP OF LATENT AND SENSIBLE HEAT. When steam is used for evaporating water from a product, the original heat in the steam is split in two ; (1) Latent Heat of vaporisation (or condensation) which is transferred to the product being treated and causes the formation of further steam by evaporating water from the product, while condensing the original steam ; (2) Sensible or water heat which is retained in the condensate which forms when the original steam condenses. The proportions of these divisions depend on the pressure at which the original steam gives up its latent heat.

The heat in the condensate at any particular pressure is greater than the heat in condensate at lower pressure. If, therefore, the pressure on the condensate is reduced, the excess heat causes some steam to be generated as " Flash " or " Self-evaporation ". Such flash steam is just as useful as any other steam at the same pressure. Its amount under various conditions is shown in Tables IV and V.

In evaporation by steam-heat, only the latent heat is transferred to the product to be treated. This transferred heat is absorbed (*a*) as sensible heat to raise the product to its boiling point and (*b*) as latent heat for vaporising water from the heated product. This water vapour from the product is at a lower pressure than the original steam, but it holds all the heat in the original steam less the sensible or water heat left behind in the condensate and any sensible heat that has been used to bring the product up to boiling point.

If now there is a use for this evaporated steam at its pressure and temperature, it can be used in the same way as steam newly generated in a boiler. It can be used for process-, water-, or space heating ; or it can be used for evaporating water from another product or from the same product from which it sprang.

Such a second evaporating process must obviously be done at a pressure lower than the first evaporation and again the heat in the steam will divide itself into (1) latent heat transferred to the product and (2) sensible heat retained in the condensate.

Clearly, provided pressures and temperatures are suitable, this process of passing the latent heat on and on can be done until the temperature and pressure have become too low for practical heat transfer. But at each stage the sensible heat is retained in the condensate. It will be seen below that much of this condensate heat can be recovered as flash and put back into the cycle, but it forms one of the practical limiting factors.

This successive use of latent heat for evaporation is called the Multiple Effect Principle. Evaporation in Single, Double and Triple Effect will now be considered. For simplicity certain important assumptions will be made :—

(*a*) The plant will be considered to be 100 per cent. efficient, i.e., no radiation or other losses will be allowed for. (See Section 477 for a brief discussion of losses.)

(*b*) The processed material will be considered to have the physical properties of water. This eliminates complications introduced by varying Specific Heat and Boiling Point Elevation.

(*c*) Only the liquid that is to be evaporated will be considered, not the concentrated result of evaporation.

(*d*) It is assumed that the object is to evaporate 1 lb. of water.

(*e*) In Figures 280 to 288 round brackets are used to indicate (Total Heat) and square brackets to indicate [Latent Heat].

471. SINGLE EFFECT. Fig. 280 shows straight evaporation in single effect under atmospheric pressure. The quantities and heat distribution are shown. Each top figure is the weight of steam or liquid in pounds. The next figures are gauge pressures and temperatures. The figures in brackets are heat contents in Btu. In this first example the method of ascertaining the heat and weight distribution will be described in detail.

The heat input is in the form of saturated steam at 54 psi gauge. (This pressure has been selected because the temperature drop between this and atmospheric pressure is divisible by two and by three into round numbers). One pound of steam at 54 psi has a temperature of 302° F., contains [909] Btu of latent heat and (1181) Btu of total heat. During heat transfer a constant pressure exists inside the heating surface (shown in the diagrams as a coil) so that, if the coil is perfectly drained, only the latent heat in the heating steam is transferred to the liquid being evaporated.

Now 1 lb. of water is to be evaporated and it enters the plant at 60° F.

It must first be raised to 212° F. requiring 152 Btu.

It then needs [971] Btu to evaporate it—a total of (1,123) Btu.

As only the latent heat of the 54 psi steam is used it will be necessary to bring in [1,123] Btu of latent heat.

As the latent or heating heat of 1 lb. of 54 psi steam is [909] it will be necessary to bring in $\frac{1{,}123}{909} = 1 \cdot 235$ lb. of steam.

This will contain $1 \cdot 235 \times 1{,}180 = (1{,}456)$ Btu of total heat.

The condensate passing through the trap will contain the total heat less the latent heat which has been given up, $1{,}456 - 1{,}123 = 333$ Btu. This is the sensible heat.

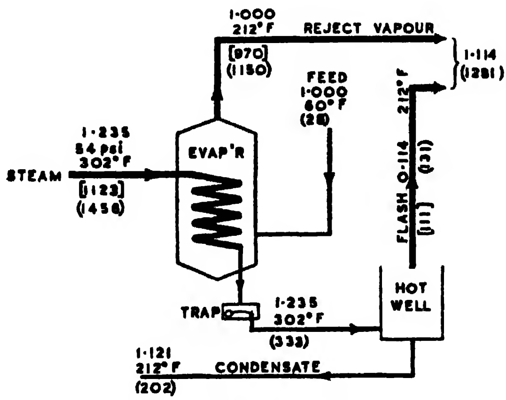

FIG. 280. SINGLE EFFECT EVAPORATOR
PERFORMANCE EFFICIENCY: 90 PER CENT.

When this condensate reaches the hot-well or condensate tank, at atmospheric pressure, the excess heat will cause a flash of steam.

One pound of water at 212° F. contains (180) Btu so that $1 \cdot 235$ lb. of water at 212° F. will contain (222) Btu.

The surplus heat in the condensate is therefore $333 - 222 = [111]$ Btu. which will cause flash.

To vaporise 1 lb. of water at 212° F. takes [971] Btu.

So [111] Btu will vaporise $\frac{111}{971} = \cdot 114$ lb.

Steam at 212° F. contains (1,151) Btu of total heat per lb.

So that $\cdot 114$ lb. will carry away $1{,}151 \times \cdot 114 = (131)$ Btu.

The condensate returning to the process weighs $1 \cdot 235 - \cdot 114 = 1 \cdot 121$ lb. and contains $333 - 131 = (202)$ Btu.

The work to be done in the evaporator is the transference of the energy needed to heat the liquid to 212° F. and then to vaporise it at atmospheric pressure, or the total heat in the vapour less the sensible heat in the feed.

This requires (1,151) Btu of which (28) is already present in the feed.

The heat taken in in the steam was (1,456) Btu.

The returned condensate contained (202) Btu so that the net heat used was 1,456 − 202 = 1,254 Btu plus the heat in the feed, which tallies with the heat rejected.

$$1{,}254 + 28 = 1{,}282 = 1{,}151 + 131.$$

So the efficiency can be said to be

$$\frac{(1{,}151 - 28)\ 100}{1{,}254} = 90 \text{ per cent.}$$

472. MULTIPLE EFFECT. The Multiple use of latent heat for evaporation can be done by closing in the evaporator and taking the vapour, previously rejected, as heating medium for use in a second evaporator. If the second vessel is to boil under atmospheric pressure the steam supplied to it as vapour off the first vessel must be under pressure. The two vessels will reach an equilibrium automatically. Assume the second vessel is open to the atmosphere, then, when steam is first admitted to the first vessel's heating surface, boiling will be vigorous in the first body and non-existent in the second body. Pressure will build up in the first body and will thus reduce the temperature drop between the input steam and the output vapour and the evaporation will slow down. At the same time a pressure and temperature difference is building up between the heating vapour and the liquid in the second vessel and evaporation will start with, of course, condensation of the first-effect vapour on the second-effect heating surface. If the rate of evaporation should rise in the first vessel its body pressure will rise and automatically reduce its rate of evaporation with an equivalent temporary increase in the rate of evaporation in the second vessel. Thus each vessel controls the rate of evaporation in the other and the self-regulation is perfect. In the examples immediately following it is assumed that equilibrium conditions will be such that there will be an equal temperature drop across each vessel. This is not necessarily the case in practice. It depends on the heat transfer rate and the sizes of the heating surfaces. The higher the temperature the higher the rate of heat transfer. The more concentrated the liquid being evaporated the lower the rate of heat transfer. In some quadruple effect evaporators the temperature drop in equilibrium in the last effect is nearly four times what it is in the first effect. The question is discussed again later. This assumption of equal temperature drops is quite fair for illustrating the principle and for driving home the technique because the various heating surfaces can be so dimensioned as to secure any temperature drop desired in any effect.

473. DOUBLE EFFECT. Fig. 281 shows an elementary double effect arrangement. The total temperature drop, that is from initial virgin steam to vapour discharged from the second effect, is 302 − 212 = 90° F., so that the temperature in the first body and in the second heating surface will be 302 − 45 = 257° F. corresponding to a pressure of 19 psi. The heat distribution is shown in detail in Fig. 281 but the working out will not be given. The method is straightforward and consists of putting in 1 lb. of 54 psi steam and following the consequences through and then reducing all weights and heat contents proportionally to give a 1 lb. evaporation.

Now Fig. 281 shows that the same evaporation has been done as in Fig. 280 but that only ·680 lb. of steam has been used instead of 1·235 lb., a saving of

45 per cent. It is more correct to compare the net heat usage. In Fig. 281 it was 598 Btu against 1,282 Btu in Fig. 280, an improvement of 51 per cent.

The work to be done is the evaporation of 1 lb. of water at atmospheric pressure, or the production of 1 lb. of steam at atmospheric pressure. This calls for 1,151 Btu. There is already 28 Btu in the feed, so that the heat to be supplied is equivalent to 1,151 − 28 = 1,123 Btu. The net heat used is the total heat in the input steam 802 Btu less the heat in the returned condensate, 204. Therefore the net heat used is 802 − 204 = 598 Btu.

The performance efficiency must therefore be :—

$$\frac{1,123 \times 100}{598} = 187 \text{ per cent.}$$

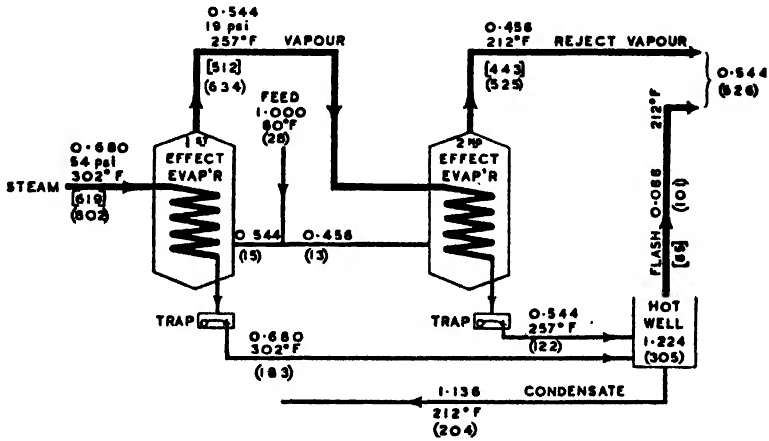

FIG. 281. DOUBLE EFFECT EVAPORATOR—PARALLEL FEED—EQUAL TEMPERATURE DROPS. PERFORMANCE EFFICIENCY : 187 PER CENT.

474. PERFORMANCE EFFICIENCY. Now we must pause to consider whether we are justified in saying that a plant can be more than 100 per cent. efficient. How are the efficiencies of different plants compared ? By expressing the work done or the energy made available for useful purposes as a percentage of the net heat or energy input.

Thus the efficiency of a turbo-generator is

$$\frac{\text{Electrical Power generated}}{\text{Heat in steam} - \text{Heat in condensate}}.$$

The efficiency of a boiler is

$$\frac{\text{Heat in steam} - \text{Heat in feed water}}{\text{Heat in coal}}.$$

It is surely rational to express the efficiency of an evaporator as

$$\frac{\text{Energy needed to heat and evaporate liquid}}{\text{Net Heat supplied to plant}}.$$

Of course the true thermodynamical efficiency of an evaporator must be zero because all the heat in the evaporated vapour is normally eventually rejected, but to use a yardstick which always measures zero is absurd. Take as an example a distillery producing, mirabile dictu, distilled water. Assume that it has first class up-to-date coal-fired stills. The management will certainly

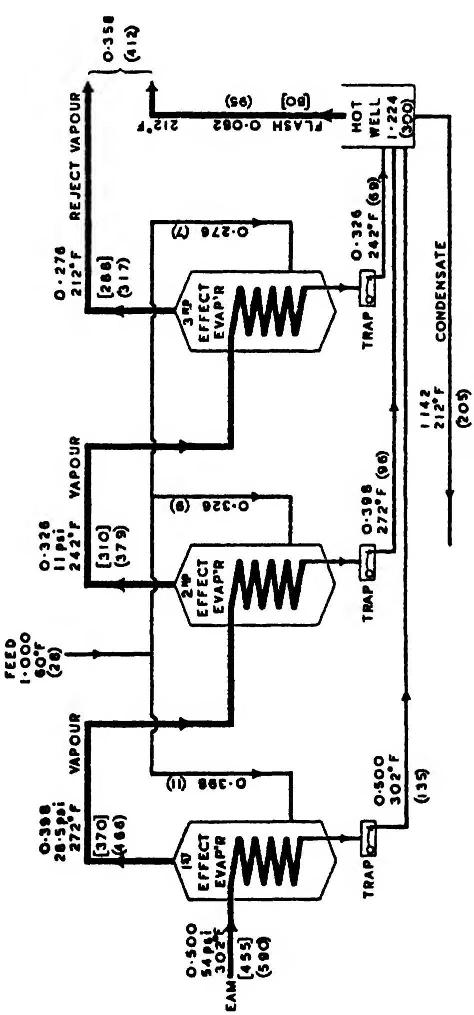

FIG. 282. TRIPLE EFFECT EVAPORATOR — EQUAL TEMPERATURE DROPS — PARALLEL FEED—PRESSURE PLANT. PERFORMANCE EFFICIENCY : 292 PER CENT.

measure the efficiency of their stills as if they were steam boilers (as indeed they are) and the efficiency may well exceed 80 per cent. Now if these stills are converted into double effect vessels it is quite obvious that the efficiency will have been approximately doubled and must exceed 100 per cent. For better or worse therefore this method of expressing performance efficiency is used in this Chapter.

Many engineers object to any efficiency figure over 100 per cent. They compromise with their consciences by dividing by 100 and calling the answer the "Coefficient of Performance". This question is discussed again in Section 768.

475. TRIPLE EFFECT. Fig. 282 shows a triple effect arrangement. The heat distribution throughout is shown. This has been ascertained by starting with 1 lb. of 54 psi steam, working through the plant and then reducing all quantities to give 1 lb. of evaporation. The total temperature drop is still $302 - 212 = 90°$ F. which must now be split into three drops of 30° F., giving a temperature and pressure of 272° F. and 28·5 psi in the first body and 242° F. and 11 psi in the second body. The total steam input has been reduced to ·500 lb., a saving of 26 per cent. on double effect and 60 per cent. on single effect. The net heat used is 385 Btu, a saving of 34 per cent. on double effect and 68 per cent. on single effect. The performance efficiency is 292 per cent

It will be noticed that from a net heat-usage point of view double effect is twice as good as single effect and triple effect is three times as good as single effect. From a steam consumption point of view, however, double effect falls well short of twice as good as single effect and triple effect is still further from being three times as good. The reason for this apparent discrepancy is that the figures of steam consumption do not take into account the heat retained by and returned with the condensate.

476. CONDENSATE. There is another benefit apart from heat economy in the use of multiple effect, namely, the extra proportion of condensate produced and available for boiler feed or process purposes. The condensate relative to the steam used is shown below :—

	Steam Used	*Condensate*	*Excess Condensate*
Single effect	1·235	1·121	— 9 per cent.
Double effect .. .	·680	1·136	+ 67 per cent.
Triple effect	·500	1·142	+ 128 per cent.

The condensate from the first vessel is pure distilled water equal in weight to the input steam so that apart from loss by flash it can look after the boiler feed. The condensate from the other vessels is distilled water, possibly slightly contaminated by the solute in the solution being evaporated. This contamination may preclude its use as boiler make-up but almost certainly is no bar to its use for process purposes.

There are many industries where distilled water for process would be very valuable. These industries may deplore their lack of distilled water, yet their plants may be peppered with reducing valves. A reducing valve can often be replaced by an evaporator which can produce distilled water for practically no cost other than capital interest. In many factories steam is raised at 60 psi and used at 10 psi. The corresponding temperature drop is 68° F. This is ample drop for a quadruple effect Still which will produce distilled water of about three times the weight of steam passed through it.

477. LOSSES. The loss by radiation and convection from a well lagged evaporator is very small. There are few published convection and radiation

loss figures. The most frequently quoted are those taken by Kerr referring to a multiple effect sugar liquor evaporator. They are :—

Number of Effects	*Per cent. Initial Steam Lost by Radiation*	
	Unlagged	*Well Lagged*
2	1·06	·26
3	4·20	1·05
4	9·80	2·70

It is important that the first effect should be well lagged because any heat lost in the first vessel is lost in all vessels. It would have done its stuff in multiple effect. The lagging can be progressively less elaborate towards the final effect.

Another loss is caused by the necessity for venting the heating surfaces to remove air or other incondensible gases. Ammonia is commonly found in sugar evaporators. Air venting is generally done by a small snifting pipe blowing from one heating surface to the next, by simply blowing into the vapour space. This loss, unlike radiation loss, is not a multiple effect loss. Any steam blown over with the air only loses one effect. But in an endeavour to make sure of adequate venting a great amount of steam is often wasted this way. It may be better to vent the heating surface by means of a thermostatic air vent device. If the pressure inside the heating surface is below atmospheric pressure, the thermostatic vent must discharge into the vapour space which will be at a lower pressure (or higher vacuum) and will therefore permit the escape of the gas.

478. NUMBER OF EFFECTS. The number of effects installed is often limited by too great a consideration of first cost. This may be short-sighted. Heating surface is one of the cheapest and most paying of investments and the number of effects should be decided on technical grounds coupled of course with financial prudence. The chief limit to the number of effects is that imposed by the Boiling Point Elevation of concentrated solutions. Thus a 70 per cent. sugar solution has a boiling point elevation of approximately 9° F. ; a 70 per cent. KOH solution has a B.P.E. of 225° F. Now the temperature drop across the heating surface must be so and so, but the vapour coming off the solution condenses in the heating surface of the next effect at the water boiling point corresponding to the pressure under which it was produced. This sets a practical limit to the number of effects that can be operated for a given total temperature drop. The limit in the sugar industry is about six effects while for evaporating caustic it is occasionally possible to use as many as three effects. The other limit is the necessity for maintaining a vigorous circulation. Good circulation greatly increases the rate of heat transfer and hence the rate of evaporation. It also prevents overheating which can occur in the bottom of an evaporator where the hydrostatic head of the liquid raises the boiling point with possible damage, especially to organic products. In the case of a water Still the snag of boiling point elevation is entirely absent. There is less increase of viscosity to slow up circulation and there is no reason why a water Still should not operate with temperature drops of 5° F. Temperature drops of 9° F. or 10° F. are not unknown in the sugar industry where boiling point elevation, viscosity and damage due to overheating of a stagnant organic product are all present dangers.

Before considering the most important part of the multiple effect technique there are a few points that must be cleared up, namely the temperatures over which the plant should operate and the distribution of the temperature drop

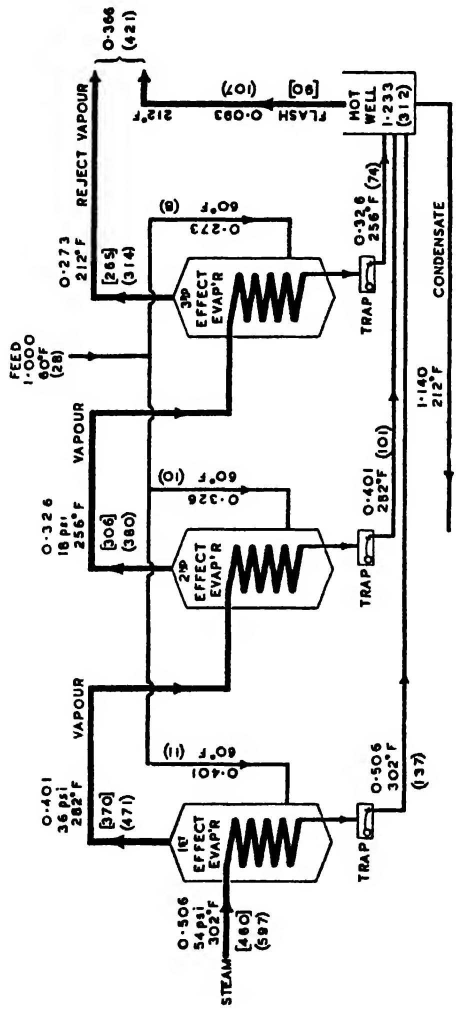

FIG. 283. TRIPLE EFFECT EVAPORATOR — EQUAL PRESSURE DROPS — PARALLEL FEED—PRESSURE PLANT. PERFORMANCE EFFICIENCY : 286 PER CENT.

479. DISTRIBUTION OF TEMPERATURE DROP. It has so far been assumed that equilibrium is reached by an equalisation of the temperature drops across each vessel. By adjusting the relative heating surface areas any desired split of the temperature drop can be obtained. This may well be desirable. At the cool end of the plant the evaporation may be too slow with the temperature drop that gives satis.actory rates of evaporation at the hot end. Let us see what the effect is of changing the proportions of the temperature drop.

In Fig. 283 it is assumed that the heating surfaces are so proportioned that equilibrium occurs with equal pressure drops. This gives the following distribution.

	Heating Surface *Pressure.*	*Heating Surface* *Temperature.*	*Body* *Pressure.*	*Body* *Temperature.*	*Temperature Difference.*
First	54 psi.	302°F.	36 psi.	282°F.	20°F.
Second	36 psi.	282°F.	18 psi.	256°F.	26°F.
Third	18 psi.	256°F.	0 psi.	212°F.	44°F.

The input steam is ·506 lb. and the performance efficiency is 286 per cent. —almost as good as in Fig. 282. The reason it is not quite so good is that a smaller proportion of the evaporation is done in the last effect, so that the application of the multiple effect principle is not quite so perfect. This is possibly unexpected. It might have been thought that a bigger temperature drop would have given greater evaporation. The reason this is not so is that only latent heat is transferred during evaporation, and the greater the temperature drop the greater the latent heat difference.

480. TEMPERATURE RANGE. The temperature range over which a multiple effect evaporator works is conditioned by several factors, the physical properties of the material being treated, for example. Thus milk or cane sugar juices call for low temperatures. Paper waste liquors, most inorganic salts and sugar beet juice can tolerate higher temperatures. The other factor of most common importance is that evaporators are almost always operated on the exhaust steam from the power producing engines, turbines or pumps. In order to keep the back pressure low the evaporator usually works over a low temperature range.

Fig. 284 shows a triple effect with the same 30° F. temperature drops as in Fig. 282 but operating between atmospheric pressure, 212 °F. and 26½ in. vacuum, 122° F. The efficiency is 360 per cent. as compared to 292 per cent. in Fig. 240, the pressure plant. The reason the efficiency is higher is because the latent heat at low temperatures represents a greater proportion of the total steam heat than at higher temperatures. As only the latent heat is transferred during evaporation, it follows that the efficiency will be higher when the latent heat represents a greater proportion. The difference between the latent heat and total heat all goes into the condensate and at high temperatures much of this heat is lost by flash. In the vacuum plant not only does less heat go into the condensate but the condensate is at such a low temperature that there is no flash at atmospheric pressure, so that all the heat in the condensate returns. If the heat and flash in the condensate could be economically used it might be possible to make the pressure plant reach the efficiency of the vacuum plant. As will be seen later the pressure plant has many thermal advantages so the various ways in which its efficiency can be improved must be investigated.

481. FEED ARRANGEMENTS. In Figs. 282, 283 and 284 each body was fed direct with its quota of liquid to be evaporated. This is called " Parallel Feed ". Let us see what happens when the feed is all put into one vessel and passed in succession to the others. When the feed is all put into the first body,

that is to say when it travels in the same direction as the heat flow it is called " Forward Feed ". When it is all put into the last body and goes against the heat flow it is called " Backward Feed ".

Figs. 285 and 286 show the heat distributions in the two feed arrangements. The method of finding the heat distribution in Fig. 285 clearly cannot be the same as hitherto. We cannot put in 1 lb. of steam and let the amount of

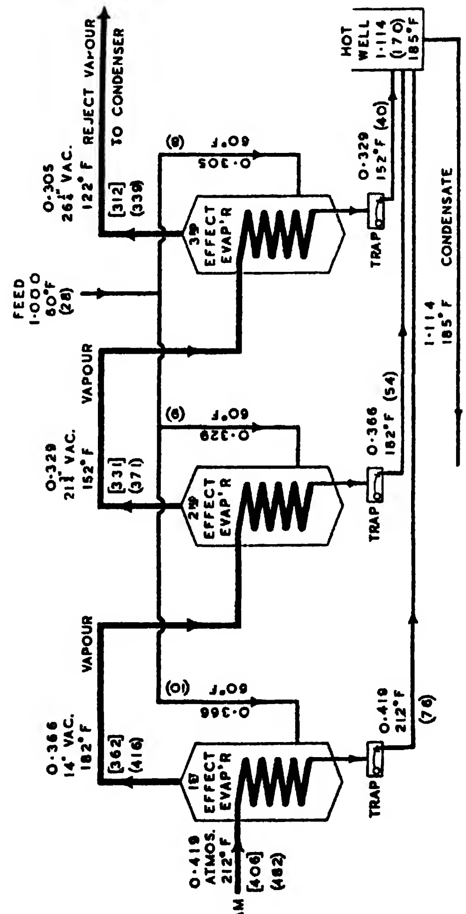

FIG. 284. TRIPLE EFFECT EVAPORATOR — EQUAL TEMPERATURE DROPS — PARALLEL FEED—VACUUM PLANT. PERFORMANCE EFFICIENCY: 360 PER CENT.

feed sort itself out. We must put in x lb. of feed. Or we can put in x lb. of steam and 1 lb. of feed. But, in fact, there is no need to take x of anything if we take 1 lb. of reject vapour and work backwards. In Fig. 286 we can take in 1 lb. of virgin steam and work forwards as before, and then reduce all the figures proportionately to give 1 lb. of evaporation.

Forward feed in Fig. 285 takes ·564 lb. of steam, uses 460 Btu and has a performance efficiency of 244 per cent. Backward feed in Fig. 286 takes ·434 lb. of steam, uses 306 Btu and has a performance efficiency of 367 per cent. Why is Parallel feed better than Forward feed and why is Backward feed better than either ? In Forward feed, Fig. 285, the whole of the feed is heated by means of

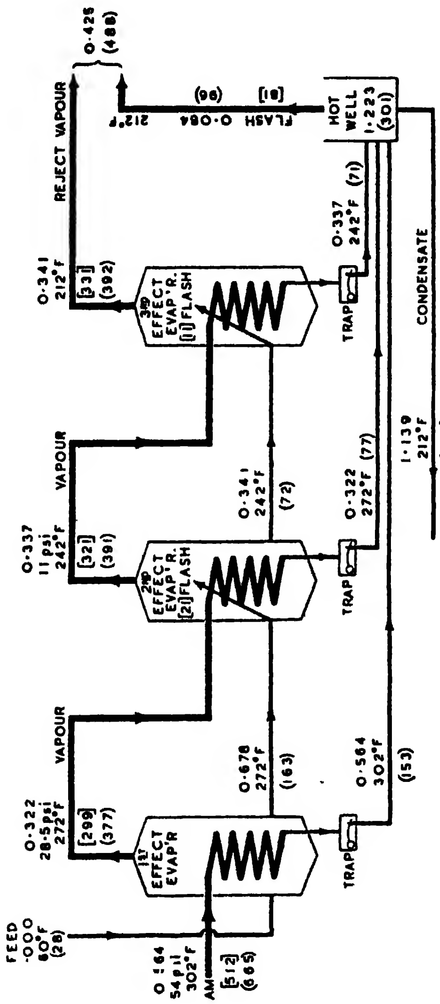

FIG. 285. TRIPLE EFFECT EVAPORATOR — EQUAL TEMPERATURE DROPS — FORWARD FEED—PRESSURE PLANT. PERFORMANCE EFFICIENCY : 244 PER CENT.

virgin steam in the first effect. There has been no re-use of latent heat for feed heating. It is also clear that all the feed has been raised to the high temperature of the first body, but this extra heat is given off as flash, or self evaporation, in the other two bodies so that the flash in the second body acts in double effect in the third vessel. In parallel feed, Fig. 282, only the ·398 lb. of feed

that is evaporated in the first body is heated up with virgin steam. The feed to the second and third bodies is heated by second hand and third hand latent heat. In the backward feed, Fig. 286, the only feed heating done by virgin team is the heating of ·410 lb. of feed from 242° F. to 272° F. The whole of

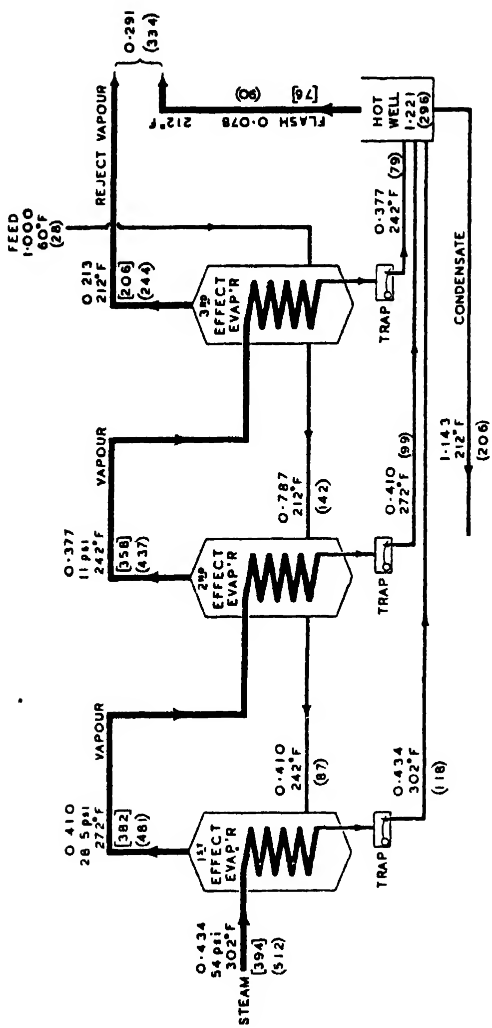

FIG. 286. TRIPLE EFFECT EVAPORATOR — EQUAL TEMPERATURE DROPS — BACKWARD FEED—PRESSURE PLANT. PERFORMANCE EFFICIENCY : 367 PER CENT.

the feed is heated to 212° F. in the third body with third hand latent heat, and the feed to the second and first bodies is heated to 242° F. by second hand heat in the second body. This is a pretty example of one of the fundamentals of heat economy technique. Use low grade latent heat for sensible heating and do it in cascade, reserving virgin heat for the final touch.

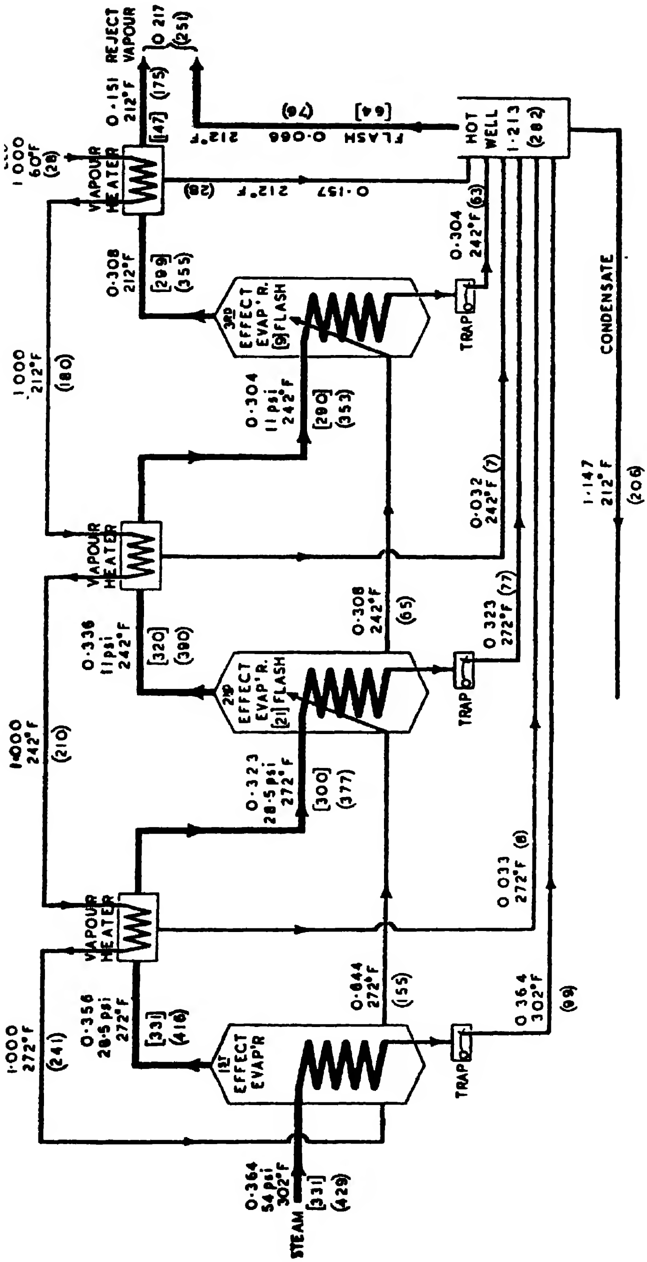

FIG. 287. TRIPLE EFFECT EVAPORATOR—EQUAL TEMPERATURE DROPS—FORWARD FEED—PRESSURE PLANT—STAGE FEED HEATING BY VAPOUR. PERFORMANCE EFFICIENCY : 504 PER CENT.

482. CHOICE OF FEED METHOD. Now it is most unfortunate that other considerations frequently compel the adoption of forward feed. For example sugar liquors can tolerate fairly high temperature for a short time in dilute solution but they rapidly deteriorate if raised to high temperature in concentrated solution. Almost every multiple effect sugar liquor evaporator works on forward feed for this reason. The other great practical benefit of forward feed is that only one feed or extraction pump is required as the feed flows naturally from vessel to vessel by reason of the pressure difference. In backward feed a pump is needed for each effect.

483. VAPOUR FEED HEATING. As forward feed is thus imposed in many cases, what can be done to improve matters ? Apply the technique just described, namely, the use of latent heat in cascade for sensible heating. This calls for a series of feed heaters in the vapour pipes. Fig. 287 shows the lay-out and heat distribution. It is clearly impossible to work out the heat distribution by simple arithmetic. It is necessary to take in 1 lb. of feed and x lb. of steam. A long, tiresome, but quite straightforward analysis will give the solution.

What has been achieved ? Something pretty wonderful. The steam has been reduced to ·364 lb., the net heat usage reduced to 223 Btu and performance efficiency is raised to 504 per cent. Why is the result so much better than backward feed ? In backward feed, Fig. 286, the first heat that the feed receives is from second effect vapour. In Fig. 287 the first heat the feed receives is from third effect vapour. In Fig. 286 the final feed heating is done by virgin steam. In Fig. 287 the final feed heating is done by first effect vapour. The feed heating in Fig. 287 is one effect better all the way through than in Fig. 286.

It may be objected that there is no temperature drop allowed for across the vapour heaters. If the liquid velocity is kept fairly high and the liquid is not too viscous, the temperature drop is not really appreciable. In cases where water is heated in such vapour heaters no appreciable temperature drop has been noticed. The conditions are not comparable with those of a turbine condenser where it is absolutely essential to effect complete condensation. Here the aim is to effect the maximum heating to the cooling medium, not the maximum cooling to the heating medium.

Fig. 287 is the exact arrangement claimed by Lillie in 1888 when he patented an early film evaporator. Lillie claimed that his vapour heaters could either be used in cascade as feed heaters or could be used for heating any convenient liquid either in stages or independently. Like Rillieux', Lillie's wonderful invention has been sadly neglected except in a few industries.

One thing more can be done to improve the heat economy. The condensate is being allowed to waste valuable potential and heat energy in flashing uselessly to atmosphere.

484. CONDENSATE HANDLING. Fig. 288 shows the method of using the flash from the condensate and of returning the condensate under pressure for boiler feed without permitting atmospheric flash. The flash from the first effect heating surface is taken from a flash pot into the vapour entering the second effect heating surface. The remaining first effect condensate and the second effect condensate are piped to a flash pot whence the flash goes into

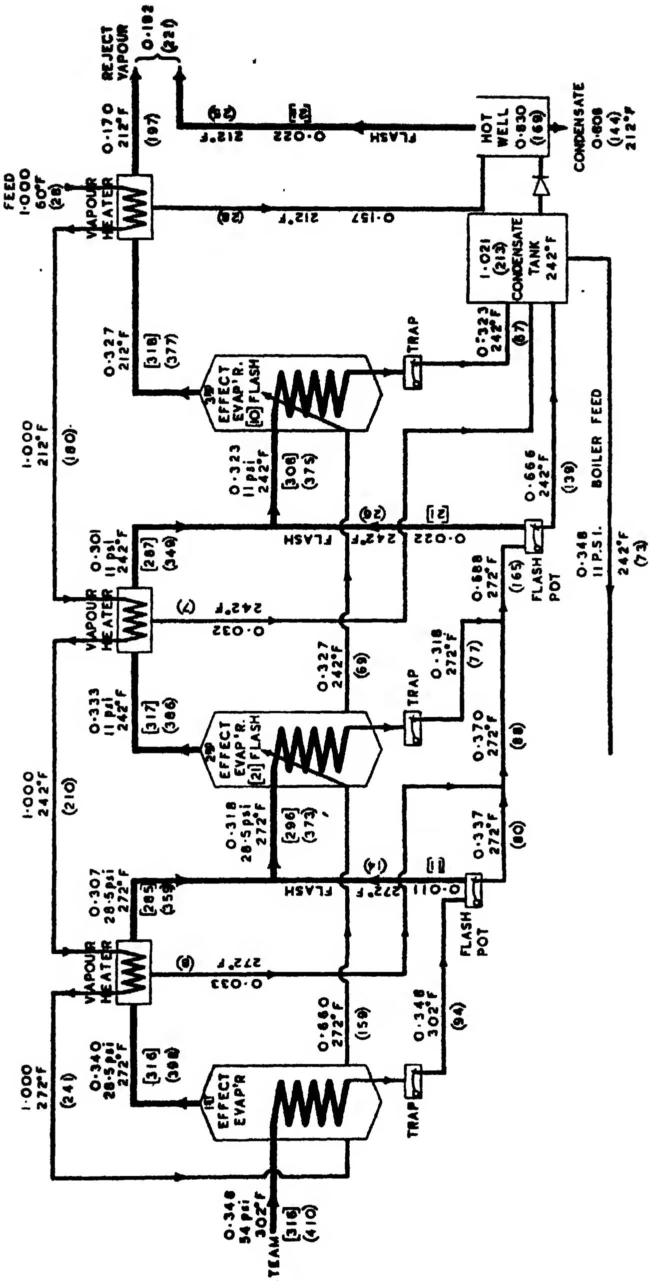

FIG. 288. TRIPLE EFFECT EVAPORATOR—EQUAL TEMPERATURE DROPS—FORWARD FEED—PRESSURE PLANT—STAGE FEED HEATING BY VAPOUR—STAGE FLASH COLLECTION—PRESSURE CONDENSATE RETURN. PERFORMANCE EFFICIENCY: 582 PER CENT.

(87664)

the vapour entering the third effect. The first flash is operating in double effect. The condensate at 11 psi is returned under pressure for boiler feed from an 11 psi closed hot well. The surplus condensate discharging into an open hot well flashes to atmosphere. The working out of the heat distribution is long, but such investigations are well worth while. The steam consumption has been reduced to ·348 lb. and the net heat usage to 193 Btu giving a performance efficiency of 582 per cent.

485. THE IMPORTANCE OF SENSIBLE HEATING. This result is so remarkable that it merits detailed study. Forward feed in Fig. 285 gave a performance efficiency of 244 per cent. This has been more than doubled by the simplest of technique, and comparatively simple plant. It must be noted that most of the increase in performance was obtained by simple hanky-panky with the sensible heating and is more than the increase obtained by going from single to double effect. Now this is the real lesson to be learnt. It is the sensible heating that really matters. No great plant is needed for correct sensible heating. It can be done, if the correct technique is applied, in hundreds of plants all over the country.

It may well be remarked that all the beautiful gain in sensible heating is due to the fact that in all the examples just considered the feed has been brought in at 60° F., whereas in practice the feed will have entered the plant hot from a previous process. How did the feed get heated in the previous process ? Answer regrettably adjudged correct—by virgin steam ! The raw water, process liquor or whatever it may be could have been passed through the evaporator vapour heaters instead of the feed. This marriage of unrelated processes is one of the most efficacious sources of steam economy.

193 Btu has been used in Fig. 288 to heat 1 lb. of liquid from 60° F. to boiling point and then evaporate it. The sensible heating accounted for 152 Btu leaving only 41 Btu to do 970 Btu worth of evaporation. This points the fundamental and valuable lesson : " Whenever evaporation can be done at a temperature high enough to allow vapour to do sensible heating, correct technique will enable the evaporation to be done almost free." This may sound a tall statement, so published authentic figures will now be given to show how very true this statement is.

486. MULTIPLE EFFECT EVAPORATION IN A BEET SUGAR FACTORY. A beet sugar factory operates through the winter on beets that may be at freezing point.

The beets are cut into shreds and the sugar is extracted by diffusion in hot water, the diffusion being carried out at about 170° F.

Taking the specific heat of beet as ·87, it will take 120 Btu to heat 1 lb. of beet to process temperature.

For every 1 lb. of beet processed, 1·25 lb. of juice leaves the diffusion battery.

This juice contains about 82 per cent., or 1·025 lb. of water.

This wintry water has to be heated to diffusion temperature, requiring 142 Btu.

The 1·25 lb. of juice, having a specific heat of say ·9, must be heated to the temperature of the first effect of the evaporator, say 240° F., calling for 79 Btu.

Of the 82 per cent. of water in the juice some 80 per cent. must be evaporated (2 per cent. going into the molasses). This requires 975 Btu.

Thus the theoretical heat requirements of a beet factory without making provision for any heat or dilution losses or for any of the many other processes is

$$120 + 142 + 79 + 975 = 1{,}316 \text{ Btu per lb. of beet.}$$

Of this, 341 Btu is sensible heating.

In 1937 particulars were published of a beet factory which was operating on the following basis :—

Coal used per cent. Beet	4·42
Calorific Value of Coal	12,450 Btu/lb.
Boiler Efficiency per cent.	70·45

This means that the heat taken into the factory from the boiler house per pound of beet was :—

$$\cdot 0442 \times \cdot 7045 \times 12{,}450 = 386 \text{ Btu.}$$

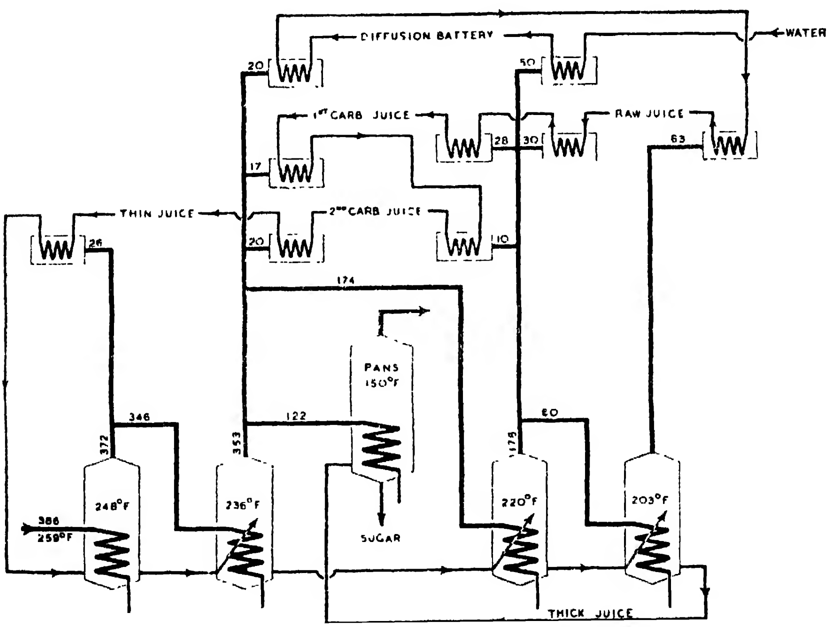

FIG. 289. THERMALLY EFFICIENT BEET SUGAR FACTORY

If 341 is the minimum " bogey " sensible heating, it only leaves 44 Btu to do 975 Btu worth of evaporation and look after all the losses. This particular factory has a quadruple effect pressure evaporator with a beautifully arranged system of cascade juice and liquor heaters, while the steam supply for the vacuum pans is taken from the vapour off the second effect of the evaporator, thus making the pans third effect vessels. The lay-out of this factory is shown in Fig. 289,

which gives the temperature and amount of steam passing to each piece of plant. This is quite a perfect example of applied heat technique and confirms the thermal slogan " Look after the sensible heating and the evaporation will take care of itself "

487. COMPARISON OF TECHNIQUE. Fig. 290 shows the same factory as is shown in Fig. 289 but without the " Sensible use of Latent Heat ". Each steam user is supplied with virgin steam. The amount of steam used is 1,366 compared with 386 in Fig. 289, or 3½ times as much. The design of an evaporator which has steam bled or passed out from each evaporator body is clearly not at all straightforward. It will be discussed very briefly in Section 493.

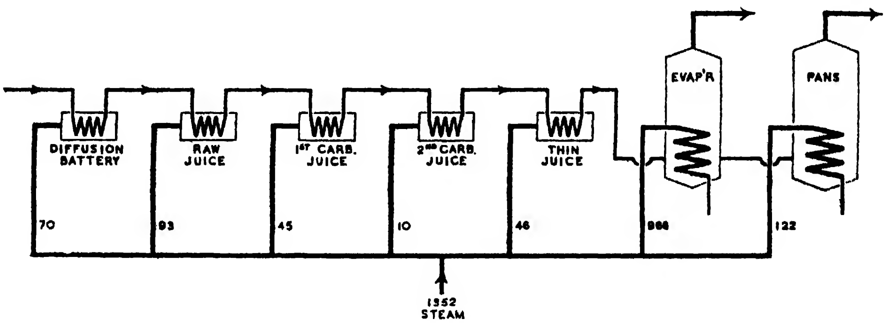

FIG. 290. HOW A BEET SUGAR FACTORY SHOULD NOT BE DESIGNED

Every industry, in fact each factory, has its own particular problems and limitations. This book cannot presume to teach any factory its business, but it can and it is hoped that it will promote and stimulate thought, interest and investigation. It may be that multiple effect evaporation is not applicable because there is no need for any evaporation. It may be that it cannot be used because of difficulties caused by scaling or boiling point elevation. But there is hardly a factory that cannot find some way of using second hand latent heat for sensible heating, and that is " in effect " simply what multiple effect technique does.

488. IMPROVED SINGLE EFFECT WORKING. The sensible heating technique that has been applied with such success to the triple effect can of course be applied to double and single effect. Fig. 291 shows what can be done with single effect. The performance efficiency has been increased from 90 per cent. to 116 per cent. by *using latent heat a second time* in what purports to be a single effect plant.

It is not necessary to describe the application of correct technique to double effect. It may be pertinent to remark here that the proud possession of a triple effect evaporator does not automatically ensure thermal efficiency. It is the correct application of the re-use of latent heat to sensible heating that brings the steam savings.

489. OPERATING TEMPERATURES. It has already been pointed out that the operating temperature range of an evaporator is frequently conditioned by the material being processed. But wherever possible the temperature range should be chosen on thermal considerations alone. If the argument has been followed thus far it will be obvious that operation over such a temperature range as to enable much sensible heating to be done by vapour is much more important than the temperature at which the final vapour is rejected. The difference in total heat in vapour at atmospheric pressure and 26 in. vacuum is only 3 per cent. Vapour at 212° F. is first class heating medium while vapour at 122° F. has poor heating value. In fact one can go further and say that in many cases, if the temperature range is reasonably high, there will be so much good sensible heating waiting to be done by the hot vapour that little or none need be rejected, but if the operating temperature is low the cool vapour is only usable for preliminary warming and much of it must be rejected.

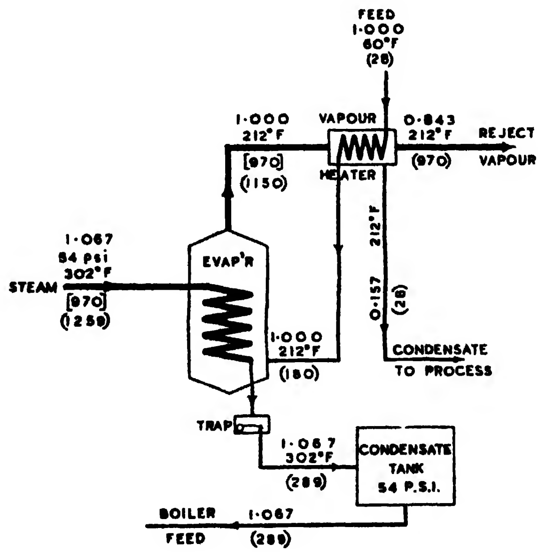

FIG. 291. SINGLE EFFECT EVAPORATOR WITH VAPOUR FEED HEATING. PERFORMANCE EFFICIENCY: 116 PER CENT.

490. POWER GENERATION COMPARED TO EVAPORATION. It is almost universal practice in process factories to generate steam at moderate pressure, to pass the steam through a back pressure turbine, engine or pump, producing the necessary power and to do any process heating or evaporation with low pressure exhaust steam, the evaporation usually being carried out under moderate or high vacuum. There is much to be said for reversing this practice. That is to say, wherever possible, doing the evaporation in pressure evaporators and using the reject vapour for power generation. There is more power energy available from steam at atmospheric pressure expanding through a turbine into a 27½ in. vacuum than there is in steam generated at 100 psi

expanding through a prime mover to exhaust to atmosphere. But the best way of generating power is " another story " and has already been dealt with in Chapters 2 and 3.

It is however important to get clear thinking as to the similarities and differences between evaporation and power generation. Evaporation by steam is the transfer of latent heat. Power generation is the conversion of sensible heat into mechanical energy with the rejection of the latent heat. (This statement is loose and any thermo-dynamist—or any zealous reader who has got as far as this—is at liberty to take exception to it, but it is quite near enough the mark to point the present argument). It follows therefore that for power generation the latent heat should be rejected at as low a temperature as possible, when the greatest heat drop has been effected. In evaporation the final rejection of the latent heat can be done at any temperature that other conditions demand. A glance at the steam table will show that it makes little difference whether the rejection temperature is moderate or very low. If proper technique is used, the reject heat will be used for sensible heating, and, if the temperatures are well chosen, there may be no latent heat to reject.

On the other hand there are marked similarities between an evaporator and a turbine. A back pressure turbine where the exhaust is used is much more thermally efficient than a condensing turbine. A back pressure evaporator is much more efficient than a condensing evaporator. The efficiency of a turbine can be greatly increased by bleeding and stage feed heating. The efficiency of an evaporator can be greatly increased by vapour bleeding and stage feed heating.

491. LIMITATIONS. The conditions, set out in Section 470, which have governed all the foregoing examples, are somewhat idealised. In practice the efficiencies given for multiple effect working will never be reached for a variety of reasons ; for example, the liquids handled industrially are never pure water and only a certain percentage of the liquor is evaporated as water vapour. Even when making distilled water some of the original feed containing the unwanted impurities is discharged as blowdown. In consequence slightly more vapour must be used for heating the feed and slightly more steam per pound of water evaporated than the ideal figures indicate.

Many industrial liquors have scaling properties. This results in it being necessary to keep the temperature drop across each effect above a certain minimum in order to obtain a reasonable output from a dirty, scaled plant. This immediately limits the number of effects that can be employed—in extreme cases only single effect is possible.

It is not usually possible to heat the feed in vapour pre-heaters to the same temperature as the heating vapour ; the liquor temperature will generally be a few degrees below the vapour temperature.

Another limitation is imposed if the liquors are corrosive. For example it may be necessary to construct the plant of nickel or stainless steel. It must be realised that a double effect is twice as big as a single effect—multiple effecting exactly multiplies the size of plant. In such circumstances financial consideration may be such as to preclude the use of multiple effect. The vapour also might be corrosive which would not only damage the plant but give unsatisfactory condensate.

492. BOILING POINT ELEVATION. Liquids with high boiling point elevations also severely restrict the scope of multiple effect evaporation. Suppose we have steam at 5 psi.g. and we are prepared to provide water and vacuum pump capacity to produce a vacuum of 26 in. We give ourselves an available temperature drop of 227 − 125 = 102° F. Suppose we decide that it is economical to provide heating surface to work with a temperature drop of about 30° F. across each effect. This means that we can operate a triple effect plant as shown in Fig. 292a, on the assumption that the liquid to be evaporated has a negligible Boiling Point Elevation.

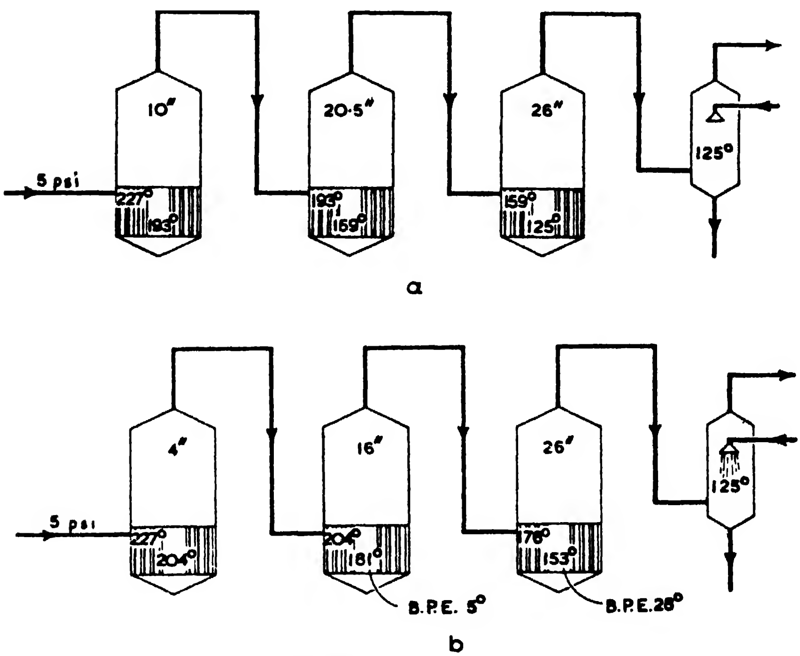

FIG. 292. EFFECT OF BOILING POINT ELEVATION

But if the liquid is to be evaporated to a considerable concentration it may have a high boiling point elevation. (For further discussion of Boiling Point Elevation see Section 778). Suppose that the concentration and physical properties are such that the B.P.E. in the last effect is 28° F. and in the second effect 5° F. We now get the conditions shown in Fig. 292b if there is an equal temperature drop over each effect.

The effective temperature drop is 102 − 28 − 5 = 69°, and the effective drop over each effect is now only 23° F.

It does not matter whether the vapour comes off the boiling liquid superheated or not. The vapour can only condense and give up its latent heat at the temperature corresponding to saturation pressure.

So that we are either forced to make the plant much larger to compensate for the reduced temperature drop or we must abandon triple effect and go to double effect.

Some solutions, such as very concentrated caustic, have such a high B.P.E. that the temperature drop of 102° F. that we have been considering, will be insufficient to overcome the boiling point elevation and such solutions could not be boiled with so small a temperature drop. For example, a 60 per cent. solution of caustic soda or a 55 per cent. solution of caustic potash each have a B.P.E. of over 102° F. so that with steam at 5 psi and a vacuum of 26 in. such solutions could not be made to boil.

These are some of the limitations that may await the uninitiated, so that expert advice should always be sought.

493. THERMO-COMPRESSION. There is another method of re-using latent heat which, although of rather limited application must not be overlooked ; this is the boosting of low pressure vapour, or exhaust steam, up to a slightly higher pressure by mixing it with high pressure steam in a suitable injector—see Fig. 293. The resulting mixed steam has a pressure higher than the original vapour and can therefore be used again. This is called thermo-compression and can be done with great efficiency when the right conditions exist. These conditions are that the steam or vapour to be boosted should be at or about atmospheric pressure, and that only a very small increase of pressure should be attempted.

Suppose an evaporator is taking 5 psi steam and is evaporating under atmospheric pressure, then 3 lb. of vapour at atmospheric pressure can be brought back to the injector, where 2 lb. of 150 psi steam will boost the 5 lb. mixture to 5 psi. In other words, where 5 lb. of vapour is being evaporated, 3 lb. can circulate round and round indefinitely. Two pounds at 150 psi are used and 2 lb. of atmospheric vapour are rejected to do 5 lb. of evaporation, which is not very far short of the performance of a triple effect evaporator. Under such circumstances plant that costs little more than single effect has nearly the same efficiency as a triple effect plant of nearly three times the cost.

Apart from the narrow limits of vapour pressure within which thermo-compression is applicable the great disadvantage is that high pressure steam must be used.

Thermo-compression may have a useful application in the evaporation of certain organic products, fruit juices, hormone extracts, etc., which are so susceptible to temperature that they cannot stand the higher temperatures called for by multiple effect evaporation, and which must therefore be evaporated in single effect at high vacuum ; but these conditions do not make for efficient working of the injector.

The other obvious application of thermo-compression is where an evaporator is fed with steam that has been blown down from a high pressure through a reducing valve. The pressure reduction should be done in an injector so that the entropy increase can give a good account of itself in doing some thermo-compression, and, with little or no cost, add the equivalent of one or perhaps two additional effects to the plant.

Thermo-compression must not be confused with steam circulation, which is discussed in the next Chapter. Steam circulation uses an injector to speed up the steam that is already in a steam space. Thermo-compression uses an

injector to draw vapour off the boiling liquid and raise its pressure to that of the steam in the steam space.

A turbo-blower can be used instead of an injector, but the power consumption is very heavy, and, unless the conditions are very favourable (the favourable conditions are the same as those for a steam injector—namely very small pressure increase and pressure around atmospheric) the cost is generally out of the question, see Fig. 294. A turbo-compressor is many times more costly than an injector and calls for skilled maintenance.

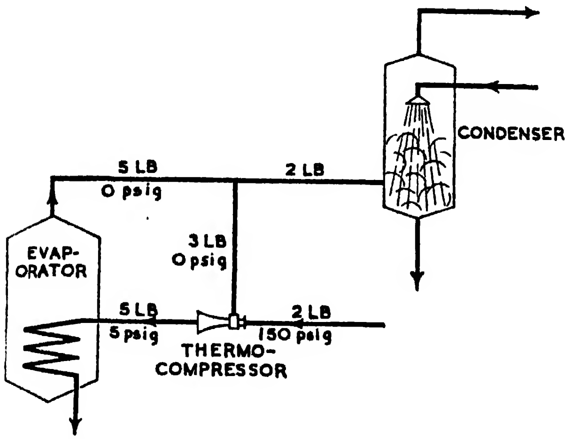

FIG. 293. THERMO-COMPRESSION WITH INJECTOR

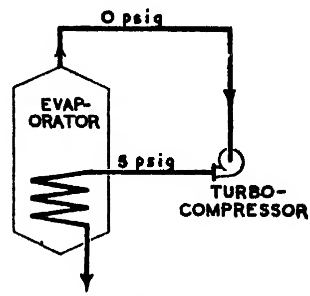

FIG. 294. THERMO-COMPRESSION WITH TURBO-COMPRESSOR

There is a distinct possibility that turbo-compressors may have a considerably improved future, because great improvements in their efficiencies have taken place in the last few years as a result of the development of the gas turbine. Some machines are stated to be nearly 90 per cent. efficient. The following example of the turbo-compression of steam has been reported :—

Amount of steam compressed ..	165,000 lb./hour
Inlet temperature	215 °F.
Outlet temperature	228 °F.
desuperheated to	221 °F.
Driving turbine steam consumption	33,000 lb./hr.
,, ,, ,, inlet ..	255 psi.
,, ,, ,, exhaust	18 psi.g.

This turbine steam consumption seems very small, but then so is the degree of recompression. The author's brother has suggested that part of the evaporator vapour be bled off to drive the compressor turbine. This should give a steam consumption equal to that in the above example and the steam can be low pressure.

494. MULTIPLE EFFECT ANALYSES AND DESIGN. Fig. 295 shows the method of analysing the heat and quantity distribution in an advanced

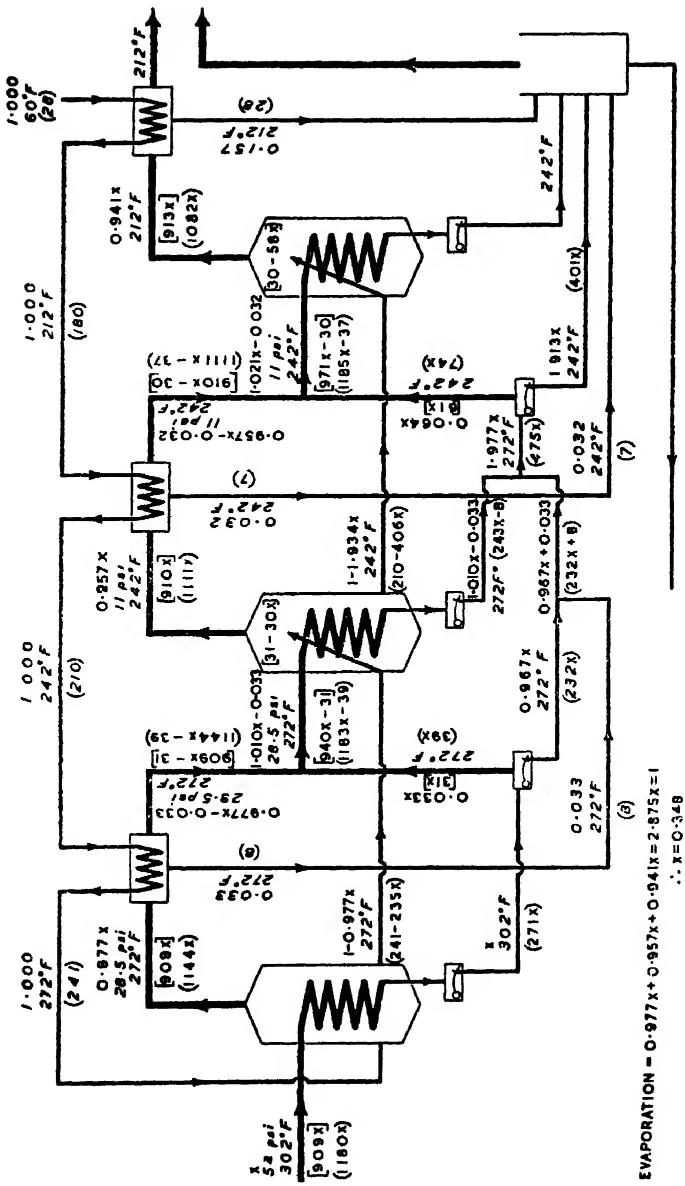

FIG. 295. ANALYSIS OF TRIPLE EFFECT EVAPORATOR

triple effect installation. The method will not be described in detail ; it can be followed from the diagram. The figures in italics are the figures that are known before the amount of steam has been ascertained and can therefore be written straight in. It should be realised that different lay-outs will require different analytical approach and no single method of attack is universally applicable.

There is however one useful hint that can be given. It is very easy in such long analyses to make a small arithmetical error which if undetected may operate " in multiple effect ". The certain prevention of such mistakes is the checking of the heat and weight balances over each piece of plant as it is dealt with.

Take the first evaporator body in Fig. 295 :—

	Weight	Heat
Input.		
Steam	x	$(1{,}180x)$
Feed	1	(241)
	$1 + x$	$(241 + 1{,}180x)$
Output		
Vapour	$\cdot977x$	$(1{,}144x)$
Feed	$1 - \cdot977x$	$(241 - 235x)$
Condensate	x	$(271x)$
	$1 + x$	$(241 + 1{,}180x)$

In the beet factory illustrated in Fig. 289 we can see that the design of a multiple effect evaporator which will do the necessary evaporation and supply the other processes with the correct amount of steam at the necessary temperatures without wasting any steam is a most complicated affair.

The first thing to do is to make an approximation. After the approximation has been made it can very easily be corrected for the things that have been ignored and omitted in the first estimate. We must of course assume that we have experience of the process that we are dealing with. Assume that each pound of beet will give us juice from which we must evaporate, in the evaporator, the equivalent of about 966 Btu of evaporation. Assume that the remaining evaporation, that must be done in the pans, is equivalent to about 120 Btu of evaporation. Assume that the sensible heating that must be supplied to the juice in its various preliminary processes is about 265 Btu. These are the figures, say, that good technique has shown to be needed with the particular process, using the particular beets in a certain part of the country.

If we draw a diagram something like Fig. 296, we can fill quite a lot of the figures in straight away. We must assume that we are going to use a pressure evaporator so that all the vapours will be hot enough to do good heating, and we will assume that we evaporate pound for pound. With proper flash collection this is quite near enough—see Fig. 288. The process requires 120 + 265 = 385 Btu/lb. beet. So, as we are going to waste nothing, this is the

amount of steam heat we must put in. Now the main steam users must draw the bulk of their supplies from the second and third bodies. The fourth is too cool for many purposes. We want to take as little first effect vapour as possible because that robs us in multiple effect. We can only use fourth effect vapour for heating the raw juice that has just come from contact with the cold beets. This heater can take about 60 Btu. The only material that need be heated by first effect vapour is the thin juice entering the first body. This heater will call for about 25 Btu. The second body must provide the pan steam 120 and we will try taking 50 additional heat units from it. The remainder 385 — 25 — 60 — 120 — 50 = 130 we must take from the third body. We therefore write these output figures in in Fig. 296. We put 385 heat units as steam into the first body.

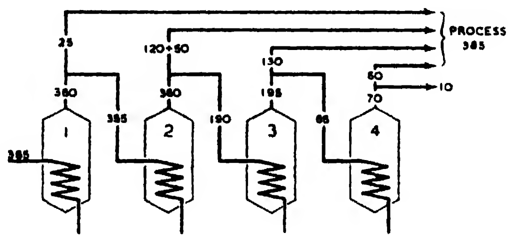

FIG. 296.

We shall lose some heat due to the lower total heat in the vapour off. So we will estimate the evaporation at 380. We are going to take 25 in the thin juice second heater so that leaves 355 to go into the second effect. We shall get flash from the first effect condensate and flash from the feed so we will assume that we get 360 off the second body. We are taking 170 of this to process leaving 190

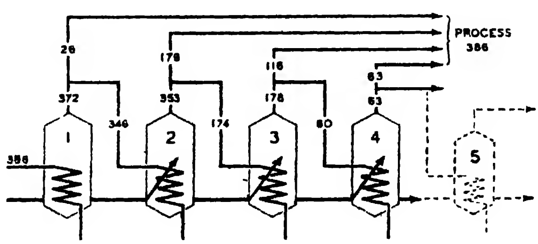

FIG. 297.

to go into the third heating surface. Assuming 195 comes off and that we take 130 to process we are left with 65 for the fourth heating surface. We will guess that with the flashes we shall get 70 off the fourth body which is 10 more than we want. What evaporation has been done ? 380 + 360 + 195 + 70 = 1,005. We have done rather too much evaporation, so we can afford to take a little more second body vapour than we have done and we are safe to make a rather more detailed estimate of the requirements. A little trial gives us Fig. 297.

As the density of the juice will vary with the season, with beet from different parts of the district and with the excellence or otherwise of our diffusion battery technique, there must be some flexibility. This can be achieved by adding a small fifth body which can be used in emergency if the juice is extra thin. Suppose we need to do 20 extra units of evaporation. With a fifth body this will operate in quintuple effect so that we shall only need to take in about an extra four of live steam and we shall only need to discard about four to the condenser. This is a very safe way of giving flexibility but it is wasteful. It is shown dotted in Fig. 297.

There is another and better way. That is to provide generous pan capacity so that in emergency some of the pans can take third body steam. Or all the pans can take third body steam part of the time. In Fig. 298 it is assumed that between a third and a quarter of the pan steam is being taken from the third body instead of the second. This gives the same heat to the process, but gives us an evaporation of 1,004 instead of 966.

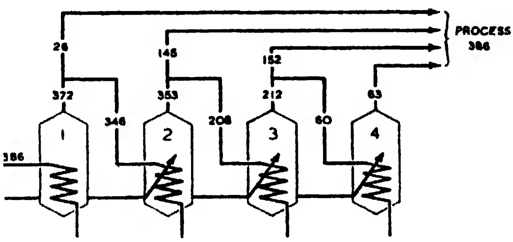

FIG. 298.

Suppose things are going very well and that the juice is fine and thick and does not need so much evaporation. We can transfer some of the juice heaters from third to second body vapour. Fig. 299 shows the effect of transferring 50 units of heating. The same steam is passed to the process but the evaporation is reduced to 916.

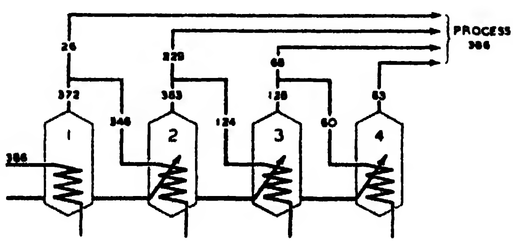

FIG. 299.

Another way is shown in Fig. 300. A bye-pass is provided between the second and third vapour mains and 50 units of steam are bye-passed. By providing such bye-passes between each effect great flexibility is secured.

Of course there are all kinds of corrections that have to be applied afterwards. All the way through the plant the specific heat of the juice is changing. All the flashes must be properly worked out. Radiation losses must be properly estimated. The foregoing little description does show the way in which these kinds of problems can be tackled without any complicated formulae or great scientific learning, but they can only be tackled approximately. The experienced expert is needed to produce the finished scheme.

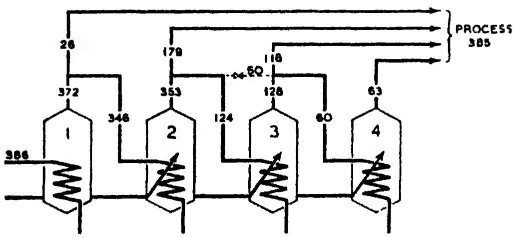

FIG. 300.

495. MULTIPLE EFFECT BLOWDOWN EVAPORATOR. The following two examples are given to show possibilities and to show what remarkable results may be obtainable. Such applications in practice may be very limited.

In Section 400, Chapter 14, a description of a compound flash heat recovery from high pressure blow down was described. It was shown that from 1,000 lb./hour of blowdown at 650 psi we could recover :—

Steam	221 lb./hr. at 70 psi
Steam	64 lb./hr. at 10 psi
Hot water	187 lb./hr. at 180° F.

Leaving only 695 lb./hr. to go down the drain at 212° F. and carrying away only 26 per cent. of the original heat. We recovered 74 per cent. of the heat with a very small simple plant.

Now let us apply, not only compound flash, but multiple effect use of this flash as well. Let us say that we will flash down to about 100 psi and put this flash steam into a multiple effect evaporator for evaporating the remaining blowdown. We will use a pressure evaporator and discharge the last effect into the 10 psi process main. We will then carry out further compound flashing and see what heat we can recover more or less regardless of practical considerations.

Fig. 301 shows an arrangement. The figures are based on 1·0 lb. blowdown and can be multiplied as requisite. It has been assumed that the heating surfaces are so proportioned that there will be a temperature drop of 35° F. across

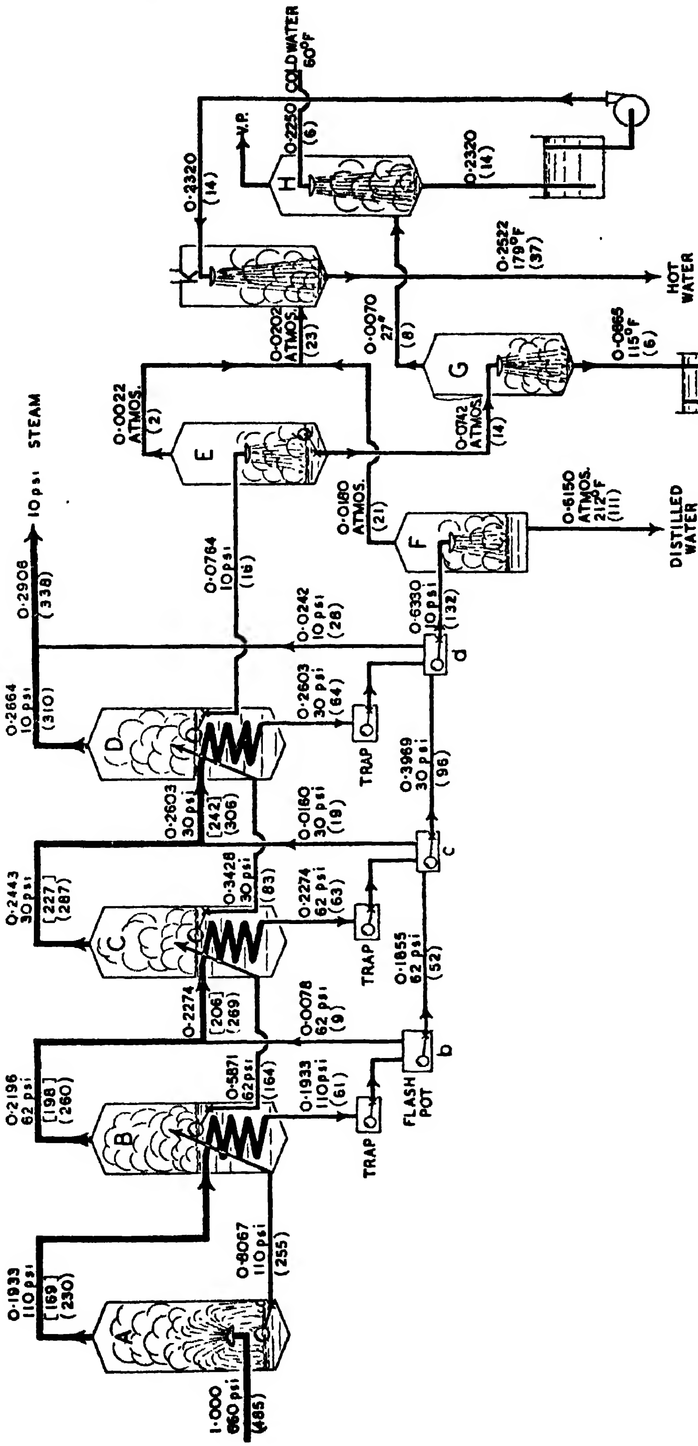

FIG. 301. ANALYSIS OF FOUR STAGE FLASH TRIPLE EFFECT EVAPORATOR ON BLOW-DOWN

each evaporator. Working back from the 10 psi output from vessel D we get :—

Vessel		*Temperature* ° *F.*	*Pressure psi.g.*
D	.. .. ..	239	10
C	.. .. ..	274	30
B	.. .. ..	309	62
A	.. .. ..	344	110

Taking each vessel in turn, we get :—

Vessel A is a plain flash vessel into which the continuous blowdown at 650 psi is sprayed. The inverted float valve acts as a trap (it can equally well be replaced by a trap), to prevent the vessel emptying completely and blowing steam.

1 lb. of water at 650 psi contains	..	485	Btu of sensible heat.
1 lb. of water at 110 psi contains	..	315·5	Btu of sensible heat.
1 lb. of steam at 110 psi contains	..	876·5	Btu of latent heat.
1 lb. of steam at 110 psi contains	..	1,192	Btu of total heat.

The surplus sensible heat in the blowdown at 650 psi entering vessel A at 110 psi is

$$485 - 315{\cdot}5 = [169{\cdot}5]$$

The flash will be $$\frac{169{\cdot}5}{876{\cdot}5} = {\cdot}1933 \text{ lb.}$$

containing $0{\cdot}1933 \times 1{,}192 = (230)$ Btu.

The feed from A to B will be $1{\cdot}0 - {\cdot}1933 = {\cdot}8067$ lb.

containing $485 - 230 = (255)$ Btu.

Check	Input to A *lb.*	Input to A *Btu*	Output from A *lb.*	Output from A *Btu*
	1·0	(485)	·1933	(230)
			·8067	(255)
			1·0	(485)

Vessel B is the first body of a multiple effect evaporator taking the flash steam from A at 110 psi into its heating surface (shown diagrammatically as a coil—it should probably be a calandria). To give a 35° F. temperature drop across the heating surface will call for a pressure in the body of 62 psi.

Feed at 110 psi has a sensible heat	315·5	Btu/lb.
Water at 62 psi has a sensible heat	279	Btu/lb.
Steam at 62 psi has a latent heat	904	Btu/lb.
Steam at 62 psi has a total heat	1,183	Btu/lb.

The surplus sensible heat in the feed is

$$315{\cdot}5 - 279 = 36{\cdot}5 \text{ Btu/lb.}$$

Flash will be $\dfrac{36{\cdot}5}{904} = {\cdot}0404$ lb./lb.

Flash from ·8067 will be $\cdot0404 \times \cdot8067 = \cdot0326$ lb.

containing $\cdot0326 \times 1{,}183 = (39)$ Btu.

Input steam from A contains [169] latent heat

This will evaporate $\dfrac{169}{904} = \cdot1870$ lb.

containing $\cdot1870 \times 1{,}183 = (221)$ Btu.

Total steam from B at 62 psi is

$$\cdot1870 + \cdot0326 = \cdot2196 \text{ lb.}$$

containing $221 + 39 = (260)$ Btu total heat.

and $\cdot2196 \times 904 = [198]$ Btu latent heat.

Feed from B to C will be

$$\cdot8067 - \cdot2196 = \cdot5871 \text{ lb.}$$

containing $\cdot5871 \times 279 = (164)$ Btu.

Condensate will be ·1933 lb.

containing $(230) - [169] = (61)$

In the flash pot *b* ·0404 lb./lb. flashes.

From ·1933 lb. the flash

will be $\cdot1933 \times \cdot0404 = \cdot0078$ lb.

containing $\cdot0078 \times 1183 = (9)$ Btu.

Condensate at 62 psi

will be $\cdot1933 - \cdot0078 = \cdot1855$ lb.

containing $61 - 9 = (52)$ Btu.

Total steam to C is $\cdot2196 + \cdot0078 = \cdot2274$ lb.

containing $260 + 9 = (269)$ Btu.

Check

Input to B *lb.*	Input to B *Btu.*	Output from B *lb.*	Output from B *Btu.*
·8067	255	·2196	260
·1933	230	·5871	164
		·1933	61
1·0000	485	1·0000	485

Input to *b*		Output from *b*	
·1933	61	·0078	9
		·1855	52
		·1933	61

Vessels C, D, *c* and *d* are calculated similarly.

We get from D and d an output of	·2906 lb. of steam at 10 psi.
	·6330 lb. of condensate at 10 psi.
	·0764 lb. of blowdown at 10 psi.
	1·0000

The 10 psi steam is useful, but there is still a lot of heat in the condensate and blowdown which we must not lose. The condensate is allowed to flash to atmospheric pressure in vessel F giving off ·018 lb. of flash steam at 0 psi.g. The blowdown is flashed to atmospheric pressure in flash vessel E where it gives off ·0022 lb. of flash steam at 0 psi.g.

These two lots of flash totalling ·0202 lb. go into the spray condenser K where they put their heat into water.

The blowdown flash vessel E is fitted with a trap or an inverted float valve to prevent its running empty. The flashed blowdown passes through the float valve into the flash vessel G connected to the jet condenser H.

In vessel G the blowdown is flashed to 115° F. giving up ·0070 lb. vapour at 27 in. vacuum. This vapour is condensed in the condenser H by ·225 lb. of water at 60° F.

The warm water from H passes through the barometric leg and atmospheric tank and is pumped into the spray condenser K where it condenses the atmospheric pressure flash from vessels E and F emerging as ·2522 lb. of water at 179° F.

The blowdown, now reduced to ·0665 lb. and 115° F., leaves vessel G, through a barometric leg and atmospheric tank, to drain and carries only (6) Btu.

We can now take an overall balance multiplying by 1,000.

Input

1,000 lb. blowdown at	650 psi containing	485,000 Btu.
225 lb. cold water at	60 °F. containing	6,000 Btu.
1,225 lb.		491,000 Btu.

Output

290·6 lb. of steam at	10 psi containing	338,000 Btu.
615 lb. of distilled water at	212 °F. containing	111,000 Btu.
252·2 lb. of hot water at	179 °F. containing	37,000 Btu.
66·5 lb. of blowdown at	115 °F. containing	6,000 Btu.
1224·3 lb.		492,000 Btu.

(By using round figures we have introduced a few small errors.)

We have recovered nearly 99 per cent. of the heat in the blowdown and we have made over 60 per cent. of make-up distilled water needed to compensate for the blowdown.

496. IMPROVED MULTIPLE EFFECT BLOWDOWN EVAPORATOR. The arrangement shown in Fig. 301 is complicated and it was arbitrarily assumed that we should have a temperature drop of 35° F. across each surface. Vessels G and H only recover a theoretical 8 Btu. With radiation loss it would certainly not be worth while putting them in.

Let us approach the problem more realistically. Every vessel must pull its weight. We must use our knowledge to design the best possible plant for our purpose. Let us assume that in addition to heat recovery our principal aim is the production of distilled water for make-up. We will also assume that the boilers whose blowdown we are going to deal with are in a power station which is working a three-bleed system where the last bleed is at 18 in. vacuum.

We know, from a comparison of Fig. 283 with 284, that we get a better result from an evaporation point of view the lower the pressure over which the plant operates. In Section 480 we saw that the smaller the temperature drop the nearer do we get pound for pound evaporation. We know that the bigger the pressure drop in the first flash vessel the greater will be the quantity of flash steam.

We will therefore give our evaporators generous heating surfaces (they will be quite small affairs anyhow) so as to get nearly pound for pound evaporation. We will make our last vessel supply the last bleed heater at 18 in. vacuum.

We see from Fig. 301 that the heat to be transferred in each effect was something over 200 Btu per pound of blowdown.

We are going to do better than this as we are going to flash off more steam in the first vessel. We shall therefore base our calculations on a transfer of 300 Btu/lb. of blowdown.

This means that we shall have to transfer 300,000 Btu/hr. in each evaporator.

From the information given in Section 366 we can assume a heat transfer rate of 500 Btu/sq. ft./hr./° F. diff.

We shall need some 600 sq. ft./ ° F. difference.

That is 40 sq. ft. with 15° F. drop.

We must make our plant of reasonable size—toy plant is not very practical—with say 45 sq. ft. of heating surface.

We shall therefore fix the outside diameter of our evaporators at 24 in.

We will use nice big tubes. If we put in two rows of 2 in. tubes in the calandrias we find we can get in 45 tubes and an 11 in. downtake. See Fig. 303a. From Tables XLIX and L we find that

1 tube 2 in. diameter has an area of	·5236 sq. ft.	per	1 ft.	length.
45 tubes 2 in. diameter have an area of	23·56 „	„		„
1 tube 11 in. diameter has an area of	2·88 „	„		„
The calandria will have an area of	26·44 „	„		„

We shall therefore need a calandria 1·71 feet high—say 1 ft. 9 in.

It has been assumed in this example that the desired temperature drops will be obtained all through the five effects with the same heating surface in

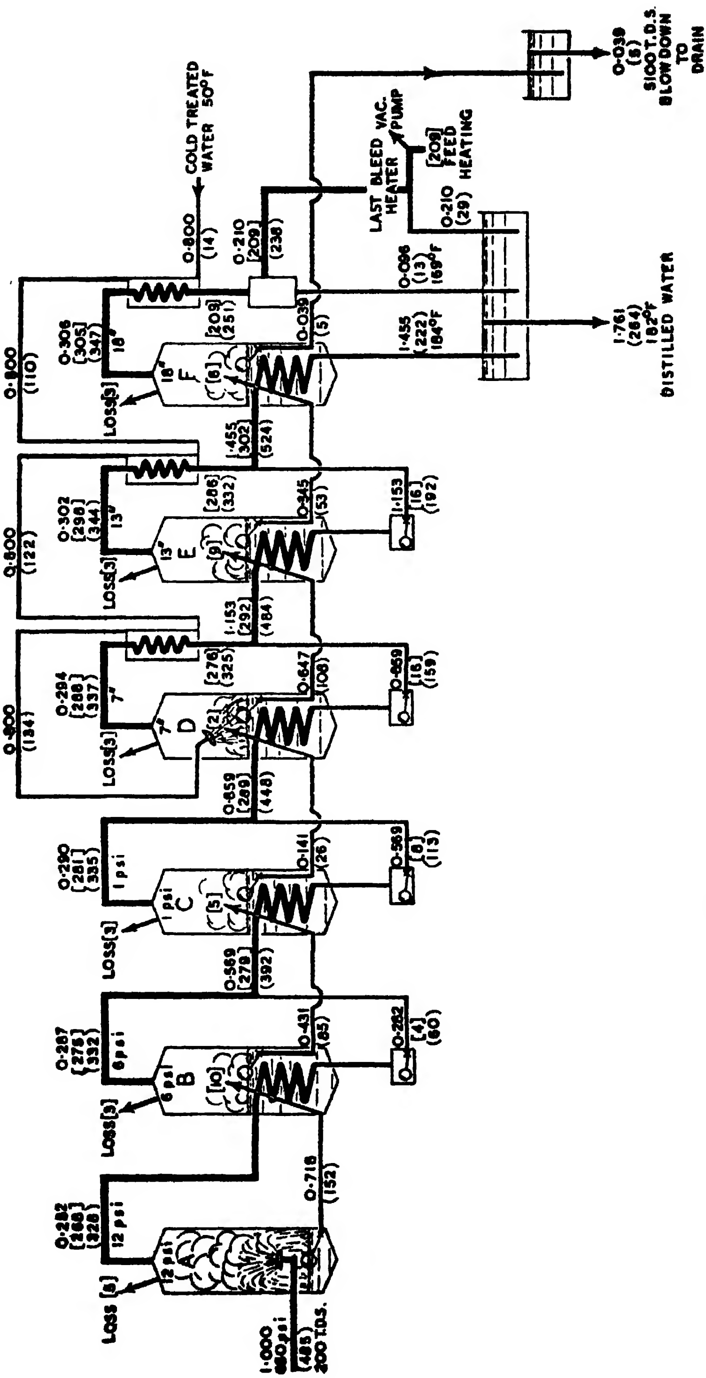

FIG. 302. ANALYSIS OF SIX STAGE FLASH QUINTUPLE EFFECT EVAPORATOR ON BLOW-DOWN

each. In order to make sure that there will be no undue falling off in performance in the cooler effects they can quite easily be given another row of 12 tubes. This would increase the total heating surface by 24 per cent. On the other hand the central downtake would have to be reduced from 11 in. to 7 in. This will reduce its heating surface by 4 per cent. So that the net increase would be 20 per cent.

At approximately 15° F. temperature drop over each body we shall get the following pressures over the plant. See Fig. 302.

Vessel			*Temperature*	*Pressure*
F	..	..	169° F.	18 in. vac.
E	..	..	184	13 in.
D	..	..	199	7 in.
C	..	..	215	1 psi.
B	..	..	230	6
A	..	..	244	12

As the heating surface is generous and we can conveniently make the plant quite roomy, we will save the complication of the flash pots by passing all the condensate through each heating surface in turn, and the flashes will take place in the steam spaces.

A loss of about 1 per cent. of the heat in the input steam to each vessel has been taken.

The feed from body C to body D in Fig. 302 is only ·141. Clearly this would be evaporated to dryness in D, leaving all the solids in the blowdown to choke the calandria in D. So we bring in enough cold water to make up the necessary feed, passing it through feed heaters in cascade as we learnt in Section 483. Most of the feed heating is in this way done with sixth-hand vapour.

We took in 1,000 lb. blowdown with 485,000 Btu.

The final blowdown only contains 5,000 Btu or 1 per cent.

We lost 20,000 Btu or 4 per cent. by radiation.

We did 209,000 Btu of feed heating.

We produced 1,761 lb. of distilled water.

A plant such as this could of course be given many more effects, and by slightly increasing the heating surface in each the temperature drop could be reduced with a corresponding increase in evaporating efficiency.

If an existing multiple effect evaporator is available the blowdown could be flashed into its heating surfaces or could be fed into the first body with the feed.

Of course if this is done the results will be very much less effective than those that have just been considered because make-up evaporators are seldom fitted with cascade feed heaters, proper flash arrangements, etc.

In the author's factory the blowdown from the high pressure boilers is blown into the first body of a quadruple effect water still and its flash acts in triple effect before going to process, while the residual blowdown is blown down with the evaporator blowdown.

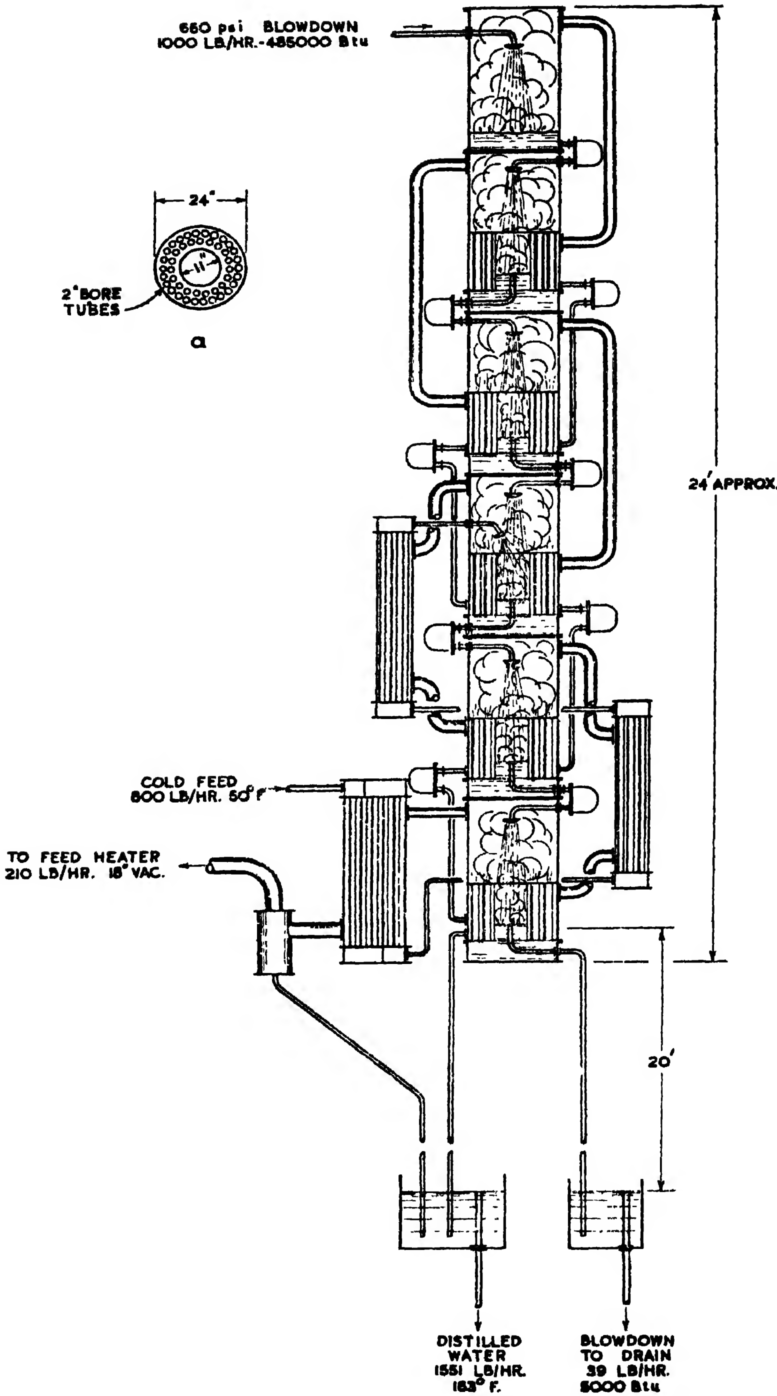

FIG. 303. DESIGN FOR THE PLANT ANALYSED IN FIG. 302

We estimated in Section 400 that the heat in 1,000 lb. of 650 psi blowdown per hour was worth 100 tons of coal per annum on three shifts. The plant analysed in Fig. 302 recovers 95 per cent. of this heat, equivalent to 95 tons of coal a year, apart from the value of 175 per cent. of distilled water. With coal at 100*s*. per ton the saving is worth about £480 a year. The plant as shown in Fig. 303 should not cost more than £1,500 to £2,000, and therefore shows a good return.

(The plant shown in Fig. 303 is imaginary—the author has never seen one like it, but there is no reason why it should not work perfectly well.)

497. SELF-REGULATION OF MULTIPLE EFFECT EVAPORATORS. Fig. 304a shows a triple effect evaporator producing distilled water while acting as a reducing valve between a steam main at 51 psi.g. and one at 10 psi.g. It is assumed that the heating surfaces are such that a 20° F. drop occurs over each body when the plant is on full load and is passing 5,000 lb. steam per hour to the 10 psi main. The output of the plant is controlled by means of a reducing valve on the outlet of the third body.

In Fig. 304b the reducing valve has closed so as to supply only half the previous steam supply. As the plant gives 5,000 lb./hr. with a 20° F. drop across each body, the temperature drops must be reduced to about 10° F. to give 2,500 lb./hr. This will and must occur automatically as has been explained in Section 472.

In Fig. 304c the reducing valve has been fitted to the input side of the first body. Again the temperature drops will even themselves out but now they will be over a lower range of pressures.

The control on the input is not desirable if the evaporator is acting primarily as a reducing valve. There is a considerable lag in response and there will be a tendency to overshoot and to hunt. If the control is on the outlet of the third body the response is immediate and each vessel can operate to some small extent as an accumulator.

498. WATERLOGGING. In Fig. 304a and b the condensate from the heating surfaces is blown by the steam pressure in the last heating surface to a height of 44 feet above the bottom of the calandria. When the control is on the outlet of the plant minimum pressure in the last calandria occurs at maximum load. When the control is on the inlet to the first effect minimum pressure on the last calandria occurs at minimum load. In Fig. 304c for example there is insufficient pressure to lift the condensate more than about 30 feet.

Fig. 305 shows a plant where the primary purpose is the production of distilled water; the plant acts as a reducing valve but this is merely incidental. The plant is controlled by the level of the water in the distilled water tank. In this case it is perhaps beneficial to control the inlet in order to get the cushioning effect of the heat capacity of the evaporator and so to reduce the fluctuations in steam output to the low pressure main that might occur were the control on the outlet. By controlling the input, the plant always works at the lowest possible pressure, whereas with the control on the outlet the pressure will be much higher—compare Figs. 304b and c. As a water Still the position

of the control is not very important. If however it is convenient to put the control on the inlet, the condensate can reach the tank 44 feet above in spite of the desire of the evaporator to settle down in an equilibrium position like that shown in Fig. 304c.

In Fig. 305 the plant is presumed to be on half load with the control on the inlet. The plant will try to work with pressure drops as in Fig. 304c.

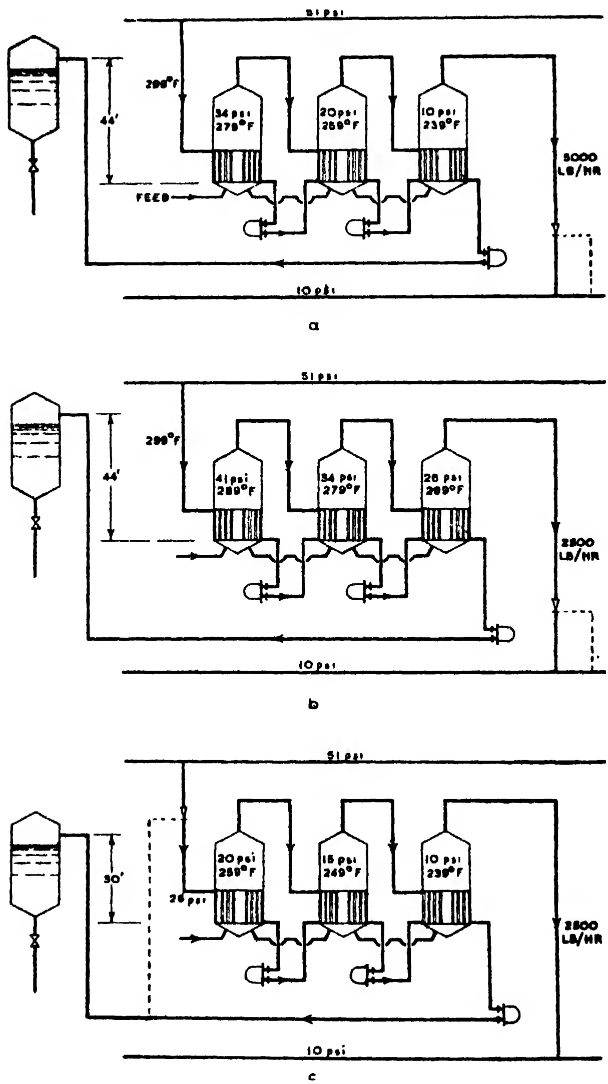

FIG. 304. SELF REGULATION OF TRIPLE EFFECT EVAPORATOR

There will be too little pressure in the last heating surface to lift the condensate so that the last calandria will waterlog. As it waterlogs its effective heating area contracts, consequently the temperature drop across the diminishing heating

surface must increase to pass the input heat. This will increase the pressure on the upstream bodies until conditions reach the state shown in Fig. 305. There is a 10° F. temperature drop across the first two bodies and there is a 20° F. drop across the last body which is half waterlogged and whose calandria is now supplied with a sufficient pressure to lift the condensate to the 44 foot level. This waterlogging control is a perfectly satisfactory way of working—it has operated in the author's factory for the last eight years.

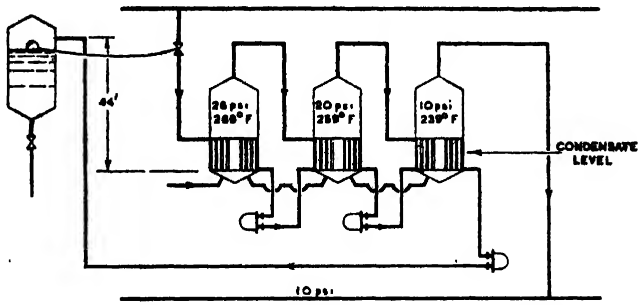

FIG. 305. WATERLOGGING CONTROL OF EVAPORATOR

Great care must be taken with the design of every waterlogging plant to minimise the risk of water hammer. No pipe should run quite horizontally. Non-return valves, if fitted, may have to be tried in several positions before trouble-free results are obtained.

499. RILLIEUX' EVAPORATOR. Fig. 306 shows the original Rillieux triple effect evaporator which was installed in a Louisiana sugar factory in 1843. The first effect normally took exhaust steam from the engines driving the cane mill. This exhaust could be supplemented when necessary with steam from the boilers. The crystallising vacuum pans were supplied with vapour off the first body.

In view of the things one sees in many factories to-day it is almost incredible that Rillieux had such an advanced plant working over 100 years ago.

The multiple effect evaporator is dealt with in a number of text books almost all of which try to deal with the analysis by means of formulae. The author believes that this is the wrong way of going about it. By the time the right formula has been found and the equations solved it is just as quick if not quicker to go about it by simple reasoning and simple arithmetic as has been done in this chapter.

The important point about multiple effect evaporation is the proper use of the output vapour for sensible heating, the bleeding off of vapour for sensible heating after it has done several evaporations and the proper heating of the feed by cascade vapour heaters.

* * *

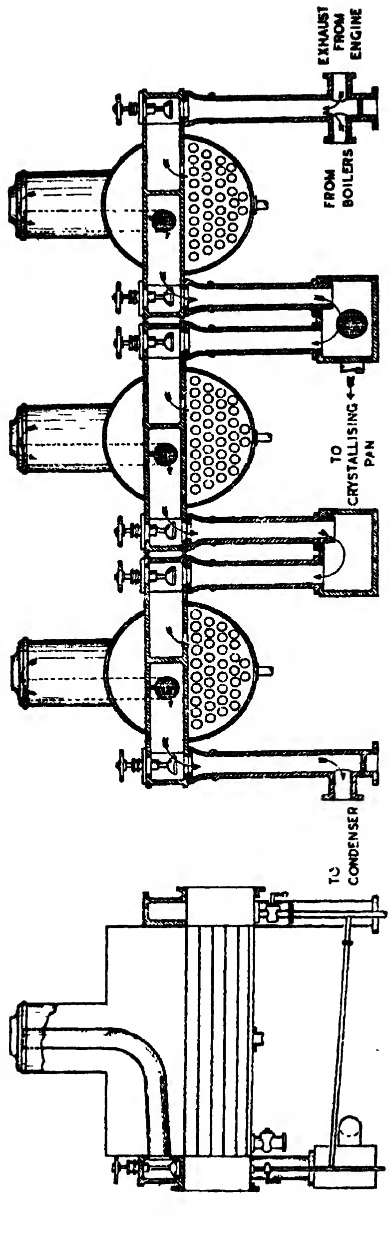

FIG. 306. RILLIEUX' MULTIPLE EFFECT EVAPORATOR—1843

CHAPTER 18

STEAM CIRCULATION AND PRESSURE HOT WATER

And to thy speed add wings.
MILTON. *Paradise Lost.* 1663.

WHERE the flow of steam inside a heating surface is sluggish, heat transfer is often unsatisfactory. Among the plants that suffer from this trouble are :—

All kinds of drying cylinders.
The platens of presses.
Long-pipe heaters.
Some textile tentering machines, etc.

Steam circulation and pressure hot water each make their own contribution towards helping heat transfer in such plants. In some cases their use, in place of orthodox steam systems, may bring about a dramatic improvement.

500. LONG-PIPE HEATERS. Steam circulation is, as its name implies, the movement of the steam round and round a steam space at a speed greater than the natural flow due to condensation on the heating surface.

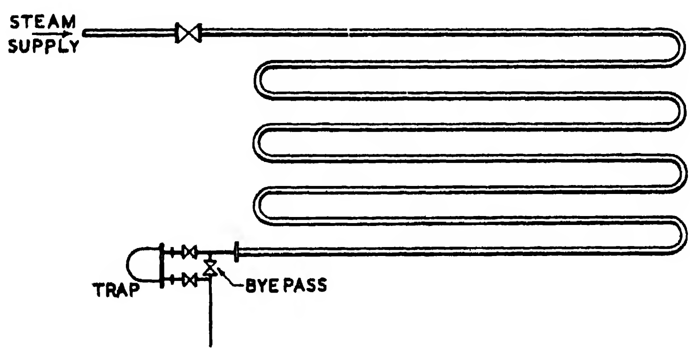

FIG. 307. LONG PIPE HEATER

Suppose we have a long steam heating coil in an oil tank, shown in the notional diagram Fig. 307. As oil is viscous and convection currents form within its bulk with great reluctance, the heat transfer rate is very low.

Suppose the coil is a 2 in. pipe 500 feet long.

Its outside surface (Table XLIX) will have an area of about 300 sq. ft.

Let us assume that the average heat transfer rate is 5 Btu/sq. ft./hour/° F. diff.

If the steam supply is at 10 psi and the oil is at about 60° F. the total heat transfer will be $5 \times 300 \times 180$ Btu/hour.

As the latent heat of 10 psi steam is 953 the amount of steam used by the pipe will be $\dfrac{5 \times 300 \times 180}{953} = 283$ lb./hour.

The cross section of a 2 in. tube is ·0218 sq. ft. and the volume of 10 psi steam is 16·5 cu. ft./lb. so that the rate of steam flow will be

$$\frac{16{\cdot}5 \times 283}{{\cdot}0218 \times 60 \times 60} = 60 \text{ ft./sec. at the beginning of the pipe,}$$

tailing off to nothing at the end—an average of 30 ft./sec.

The last 100 ft. or so will have a negligible heat transfer and will contain mostly stagnant air and condensate.

In an endeavour to improve matters, only too often, the trap bye-pass is opened and the system is allowed to blow through to atmosphere. This generally brings about a great improvement and the trap is blamed. It is the design of the heating system that is to blame.

501. THE CIRCULATOR PRINCIPLE. Let us modify the system shown in Fig. 307 so that it looks something like Fig. 308. The steam system is made a continuous loop. At the end of the heating coil is a separator in which the air and the condensate can separate from the steam. The steam can flow freely out of the top of the separator into the circulator.

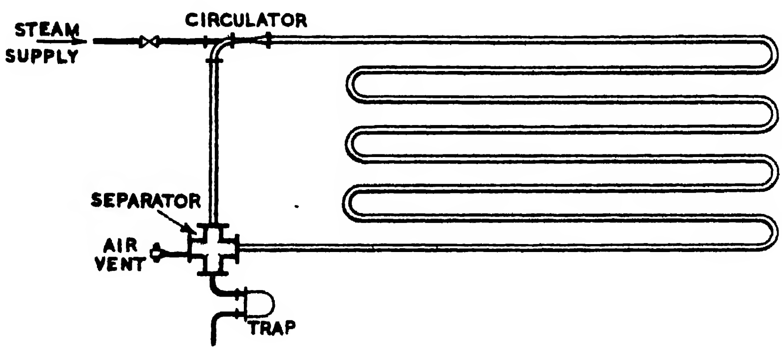

FIG. 308. LONG PIPE HEATER FITTED WITH STEAM CIRCULATION

The circulator is shown, in an idealistic sort of way, in Fig. 309. The input steam flows through a nozzle A into a venturi throat B and, by acquiring a high velocity in the throat, produces a suction in the circulating pipe C. This draws in a considerable amount of steam through C which is slightly compressed in the outlet D. In this way a pressure difference is set up across the circulator, which means that there is also a pressure difference across the long pipe. This causes the steam to flow round and round. This flow clears all the air out of the heater pipe and, by its movement, brushes the condensate along into the separator. The effectiveness of the heating surface is thus greatly increased. The temperature all through the long pipe is the steam temperature—not the temperature of cool air or tepid condensate. The result is a greater heat transfer rate, causing quicker condensation, which in turn demands a larger steam inflow. The faster inflow increases the injector action of the circulator and speeds up the circulation.

In order to induce a proper circulation in steam circulation systems, it is essential that the input steam be at a higher pressure than the steam inside the heating surface.

While steam circulation may greatly improve the performance of long pipe heaters, it is probably even better to cut the long pipe into pieces—in other words, abolish the long pipe—see Section 349.

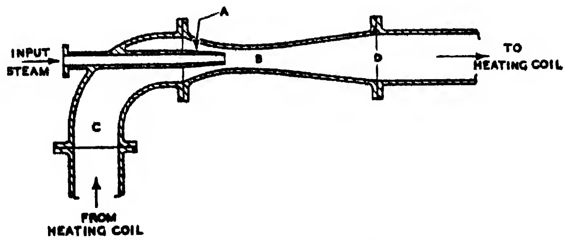

FIG. 309. DIAGRAMMATIC STEAM CIRCULATOR

502. **SEPARATOR.** The separator is most important. It would be very undesirable to carry the air round and round. Its partial pressure would gradually increase and cause a progressive lowering of the steam temperature, see Sections 318 and 404, and Table LVIII. The separator must also clear the condensate from the system.

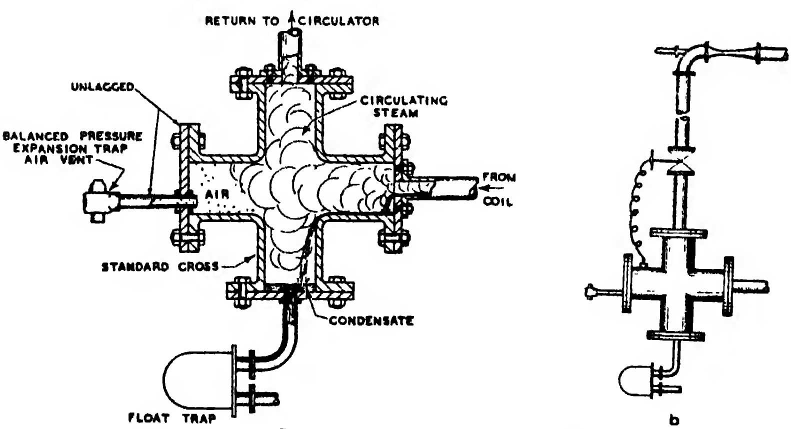

FIG. 310. AIR AND CONDENSATE SEPARATOR ON STEAM CIRCULATION SYSTEM

A large standard cross makes an excellent separator and is shown in Fig. 310a. It provides a good sump for the condensate and a fine big blind alley in which the air can collect. The condensate can be removed by any kind of trap, but preferably by a plain float trap which will remove the

condensate continuously and does not have to try to remove air which it cannot do. The air should be removed by means of a balanced pressure expansion trap put at the end of a short cooling pipe.

The arrangement shown in Figs. 308 and 310a makes no provision for quick venting of the air at the start up. Plant amenable to steam circulation is often of large volume. When the steam is first turned on, the air is driven towards the separator ; the plant is cold and initial condensation is rapid. The consequence is that the circulator becomes a true ejector and creates a vacuum in the separator and air would be drawn in through the air vent.

This can be avoided by fitting a valve between the separator and the circulator. This valve can be worked by hand or can be thermostatically operated. Fig. 310b shows the latter. The thermostat bulb is fitted in the air pocket in the separator and actuates the valve in the steam pipe leading to the circulator. So long as there is a large quantity of air in the separator the thermostat closes the valve. This ensures a pressure in the separator to discharge the air. As soon as steam reaches the separator the thermostat opens the valve which connects the separator steam outlet to the circulator suction.

503. CIRCULATORS. The circulator shown in Fig. 309 is not really suitable in practice. A fixed injector such as this will only operate properly under constant conditions of steam input, whereas we probably want to vary the steam input. Circulators are generally combined with the controlling steam valve. There are several different types.

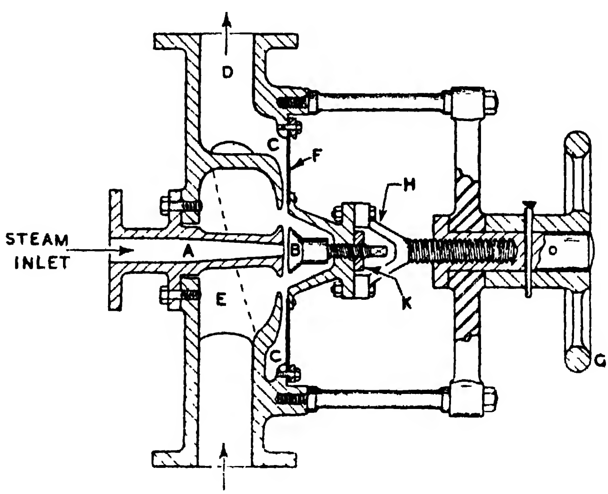

FIG. 311. ONE TYPE OF COMBINATION STEAM CIRCULATOR AND CONTROL VALVE

Fig. 311 shows one of these types. The live steam enters through the nozzle A which can be closed by valve B. The steam flows out radially as a disc and forms a radial injector into the space C, which is connected with the

outlet D. It draws steam from space E. As the valve B opens and allows more steam to pass, the annular injection throat must be allowed to get larger. This is done by diaphragm F whose centre is moved with the valve when the handwheel G is turned. In order to find the best relation between valve opening and diaphragm movement, the valve B has a threaded stem which can be screwed in or out of the yoke H and locked in the appropriate position by the lock nut K. The contours of the nozzle, the valve body and the diaphragm must be shaped to simulate an injector throat.

504. **CIRCULATION ON DRYING CYLINDERS.** Circulators have been applied to the cylinders of paper drying machines with some success, although this application is not ideal. As a paper machine almost always works at several pressures, usually three, there must be three circulators. Some fairly complicated systems are in use, some of which use the circulator as a flash ejector to remove as much flash steam as possible from the condensate before it is discharged. There are no traps on the individual cylinders, the only traps being on the separators. The steam may circulate round the easiest path. It is hardly practical to provide a circulator to each cylinder, though this is what the plant calls for. Some of the paper machine circulators are not really circulating systems, but are series systems with flash ejectors.

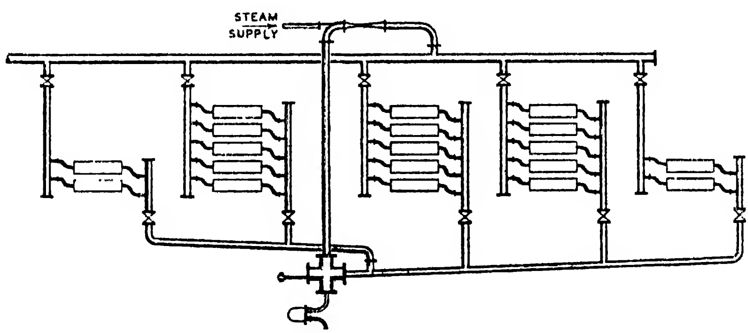

FIG. 312. STEAM CIRCULATION APPLIED TO A BATTERY OF MULTI-PLATEN PRESSES

505. **CIRCULATION IN PRESSES.** Fig. 312 shows a battery of presses connected to a single circulating loop. This is possibly better than fitting each platen with its own trap—it is probably better than fitting one trap in each press—but, like the paper machine, it suffers from the fact that those presses whose steam pipes and platens have the least resistance to steam flow will steal all the circulation and do much better than their less fortunate neighbours. The uncertainty of steam circulation is more marked with multi-platen presses where it is out of the question to fit each platen with its own circulator.

506. **SPEED OF CIRCULATION.** It is well nigh impossible to predict the speed of circulation of the steam in a circulation system. It depends on

the amount of input steam, its pressure drop in the circulator, the design of the circulator, but chiefly on the resistance to circulation offered by the piping and steam spaces.

The actual speed of circulation cannot be measured by orifice because the introduction of an orifice of sufficient constriction to give a readable pressure drop would offer so much resistance to circulation that the measurement would be useless. The only possible way would be by means of a Pitot tube, see British Standard Code No. 1042 of 1943.

In a long-pipe heater the resistance is generally large, but, as has been pointed out in Section 349, the best thing to do is to cut the long pipe into two or more sections. The same applies even if circulation is to be used. If a long-pipe heater is cut in half the resistance of each half is half of the original. If the two halves are put in parallel the resistance of the two will be one quarter that of the original long pipe. This would allow nearly four times the circulation rate.

The resistance of a drying cylinder depends only on the inlet and outlet so, if cylinders were being changed over to circulation, it would be essential to ensure that the condensate outlet and dip pipe were generous in size.

The resistance offered by press platens is usually their extremely small inlet and outlet connections. It may be possible—it would certainly be desirable—to try to arrange for two inlets and two outlets to each platen.

The circulation principle only properly applies to heating surfaces of the right shape and cross section, i.e. long and thin—long pipe heaters, press platens, etc. The drying cylinder is not really very suitable. The circulated steam will not brush the surface of a drying cylinder. The reason that circulation often improves drying cylinder performance is that air is swept out into the separator and a pressure drop is created across the discharge pipe, thus minimising the evils of group trapping.

Really positive circulation can be obtained by means of mechanical blowers. These are somewhat extravagant in power but are quite positive and really do produce a circulation which can only be surmised with the ejector type.

507. PRESSURE HOT WATER. Pressure hot water may offer a solution to many difficult heating problems. It is much more definite than steam circulation, but it suffers from a serious disadvantage ; there must be a progressive temperature drop in the water and this may preclude its use for many applications.

Pressure hot water simply consists of hot water under pressure at a temperature below the appropriate boiling point. If water is put under pressure its temperature can be raised considerably before it reaches boiling point. The steam table gives the boiling temperatures corresponding to the various pressures. As it is important that there should be no chance of steam forming, the pressure must be sufficiently high to ensure that the required temperature is well below the boiling point. Care must also be taken to avoid possible pressure drops at valves, etc., which might produce flash and cause water hammer, or a singing or rattling due to a continuous flash and continuous recondensation, which may damage the valves.

508. HEAT CAPACITY OF PRESSURE HOT WATER. Steam will flow naturally into the steam space of a heating surface because the steam condenses as it gives up its heat and this causes a local pressure reduction so that the steam is " drawn " towards the heating surface.

If we have a steam heated calorifier taking steam at 20 psi, heating water from 150° F. to 180° F. with a heating surface of 50 sq. ft., the amount of steam that will flow into the calorifier can be predicted approximately.

The temperature differences are, initial 109°, final 79°. As these do not differ by twice there is no need to take the log mean.

The arithmetic mean temperature difference is $\frac{109 + 79}{2} = 94°$ F.

(The log mean temperature difference is 93·3°.)

Let us assume a heat transfer rate of 250 Btu/sq. ft./hr./° F. difference (Table XLVII).

At 20 psi saturated steam has a temperature of 259° F. and a latent heat of 940 Btu/lb.

The steam taken, if the plant is working flat out will be

$$\frac{250 \times 50 \times 94}{940} = 1{,}250 \text{ lb./hr.}$$

Now suppose we wanted to replace steam by pressure hot water (this would probably be silly with a straightforward calorifier but this is being used purely as a numerical example) we must find out how much water we must circulate and at what temperature the water must be. Let us assume a heat transfer rate, water to water, of 100. We shall therefore need an average temperature difference of two and a half times that required for steam. In addition to temperature difference we must take account of the temperature drop in the water—each Btu removed from a pound of water lowers its temperature by one degree.

The hotter the water, the greater the water temperature drop that can be permitted, but the higher must be the pressure. The cooler the water the greater must be the quantity circulated.

If the calorifier is unchanged and the heat transfer rate is reduced from 250 to 100 we must secure an average water temperature of

$$\frac{150 + 180}{2} + \frac{94 \times 250}{100} = 400° \text{ F.}$$

Let us stipulate that the water temperature must not drop more than 40° F.

The outgoing water must have a temperature of 380° F.

and the ingoing water must have a temperature of 420° F.

It must therefore have a minimum pressure of 300 psi.

Water at 420° F. contains 397 Btu/lb. and water at 380° F. contains 354 Btu. Each pound of water circulated will therefore part with 43 Btu.

When the calorifier was steam heated it absorbed

$$250 \times 50 \times 94 = 1{,}175{,}000 \text{ Btu/hour.}$$

It will be necessary to circulate 27,325 lb. of pressure hot water per hour or 45 gallons a minute.

Suppose we say that we cannot afford to pump this quantity; that we can only spare the power to pump 30 gallons per minute. In Section 334 the effect of liquid movement was discussed and it was suggested that when precise figures were lacking it could be assumed that heat transfer rate varied as the cube root of the liquid velocity. The cube root of 45 is 3·56 and of 30 is 3·11. The heat transfer rate at the lower rate of circulation will drop from 100 to

$$\frac{100 \times 3{\cdot}11}{3{\cdot}56} = 87 \text{ Btu/sq. ft./hr./}^\circ\text{ F. diff.}$$

Each pound of water at the slower rate must give up $\frac{43 \times 45}{30} = 65$ Btu and the average temperature difference across the surface must be increased

$$\text{from } 400 - 165 = 235^\circ \text{ F.} \quad \text{to} \quad \frac{235 \times 100}{87} = 270^\circ \text{ F.}$$

The average water temperature must be $270 + 165 = 435^\circ$ F.

Water at 435° F. has a sensible heat of 414 Btu/lb.

If we say that 33 Btu must be given up either side of 435° F. the pressure hot water must enter the calorifier with 447 Btu/lb. at a pressure of 475 psi and a temperature of 465° F. and must leave with 381 Btu/lb. and a temperature of 405° F.

509. PIPE LINE REQUIREMENTS. Now we see from this that, unless a very big temperature drop in the hot water is permissible, the quantities of pressure water that must be circulated are very large compared to the corresponding amount of steam that is needed—somewhere about 20 or more times as much with temperature drops of the fairly high order of 40° F. to 50° F. But water at 500 psi only occupies $\frac{1}{600}$ of the volume of steam at 20 psi. On the other hand steam can flow about 12 times faster than the same weight of water.

So we can say, roughly :—

Volume of steam	600 times that of water
Speed of steam	12 times that of water
Quantity of water	20 times that of steam

So that the steam pipe must be about $\frac{600}{12 \times 20} = 2\frac{1}{2}$ times the area of the equivalent pressure hot water pipe.

With pressure hot water we need two large pipes of equal size against one much larger steam pipe, a small condensate return, a multitude of traps and air vents and a flash collecting system.

The foregoing estimate of pipe size is not a general statement. The size of a steam pipe will depend on the pressure. The size of hot water pipes will depend on the permissible temperature drop.

510. APPLICATIONS OF PRESSURE HOT WATER. The ideal applications are to plants where the volume of the heating space is small and where heat transfer is very sluggish. An example has been given of the advantages of steam circulation in Sections 500 and 501, in long-pipe heaters. Pressure hot water gives a more certain gain but may call for more power than can be spared. Another useful application is to the platens of presses, but it is necessary to be sure that the water will sweep right over the heating surfaces. If there are cul-de-sacs and blind corners, they will be filled with almost stationary cool water which may be just as bad as stagnant air in a steam system.

Pressure hot water is seldom the first choice in any easy heating job. Steam always wins if it can be made to work, but where steam fails or is only moderately successful, pressure hot water may show a great improvement. The outstanding advantage of pressure hot water is that it can go anywhere—up—down. There is no need to find a proper falling path for the condensate return. This gives a freedom to lay-out and a tidiness to design that are very valuable.

511. SHORTCOMINGS OF PRESSURE HOT WATER. The principal disadvantage of pressure hot water is that the water must drop in temperature when it parts with heat. If exact temperatures are called for, this means that the speed of circulation must be very high indeed. This dilemma of temperature drop or vast circulation rate may put pressure hot water right out of the running. The other disadvantage is that where there are large dead spaces inside a heating surface there may be regions of low temperature.

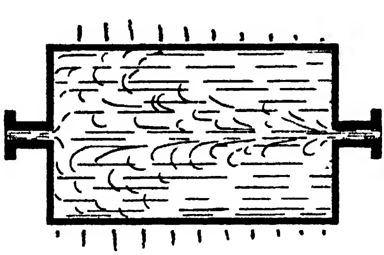

FIG. 313. LARGE VOLUME OF DRYING CYLINDER PREVENTS PRESSURE HOT WATER CIRCULATING PROPERLY

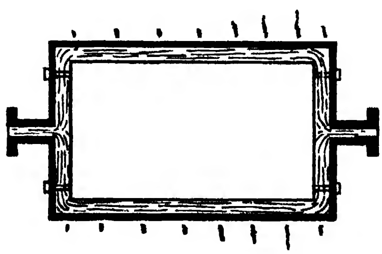

FIG. 314. BY FITTING FALSE INTERNAL CYLINDER IN DRYING CYLINDER THE HOT WATER IS GIVEN A GOOD PATH AND WEIGHT IS REDUCED, BUT WATER SUFFERS A TEMPERATURE DROP

Of all the plants where steam shows up badly the drying cylinder is one of the most important because it is so convenient and is consequently in use in many different industries, textiles, paper, milk, laundries, etc. Fig. 313 is intended to show the probable flow of water through a drying cylinder heated by pressure hot water. The chances are that no fresh hot water reaches the heating surface at all. A few warm eddies may occur near the left-hand outlet side. The things like flames in the picture are meant to represent heat transmission. Apart from this failure of the hot water to reach the heating surface, a huge load is imposed on bearings and driving gear by the great mass of water.

If an endeavour is made to combat these troubles by fitting an internal false cylinder, as in Fig. 314, another trouble appears. There will be a temperature drop across the face of the cylinder, and the thinner the water space the greater will be the temperature drop, yet a thin water space is the only one that will ensure the water reaching the heating surface and loading the machine with a minimum weight. A temperature drop across the cylinder such as is suggested in Fig. 314 would be quite out of question in, for example, paper machines.

Pressure hot water may be out of the question if the temperature and therefore the pressure has to be high, because the cylinders would have to be so heavy to be sufficiently strong.

So we see that the drying cylinder still remains the most difficult piece of plant to heat in a really satisfactory way.

512. THE PERKINS SYSTEM. One of the earliest, if not the first, systems of pressure hot water heating was the Perkins system, which is shown in Fig. 315. Such a system, used for heating lithographic stoves, has been in use in the author's factory for over 35 years with satisfactory trouble-free results.

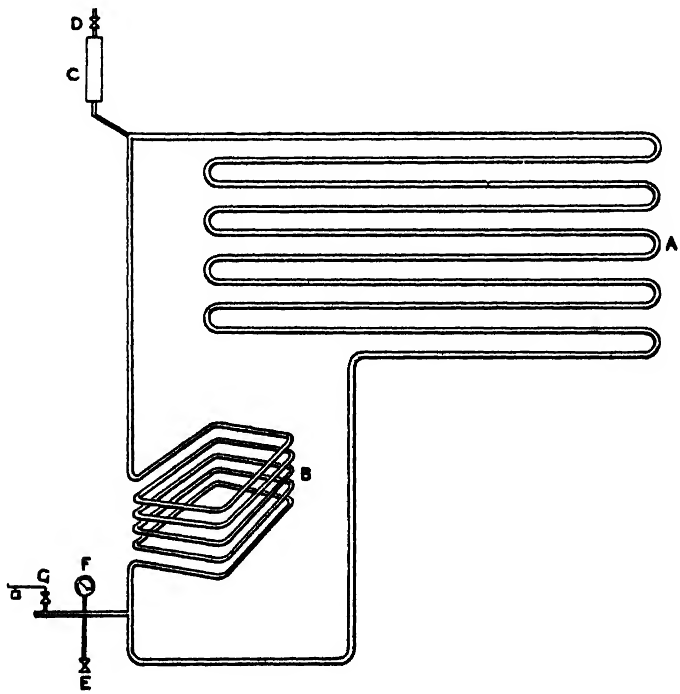

FIG. 315. THE PERKINS PRESSURE HOT WATER SYSTEM

The heating coil is a single long small-bore, about $\frac{7}{8}$ in., steel tube which if arranged in plain zig-zag coils A in the stove to be heated. The tube is wound in a close coil in a refractory box B to form the boiler.

The system works at no definite pressure; there is no method of controlling the pressure. If more heat is wanted, the boiler fire is stoked more vigorously, more heat goes into the water and the pressure goes up. As the water is heated its volume increases. To accommodate the expansion an air bottle C is fitted to the topmost part of the circuit. This bottle can, if high temperatures are desired, be precharged with compressed air. The bottle acts as a safety cushion, as well as a collecting point for any air that may be driven off the original charge of water. The cock D at the top of the bottle allows the cylinder to be charged with air, or excess air to be drawn off. The system is charged with water through the connection E. A pressure gauge F is fitted at the bottom of the circuit near the boiler to indicate to the stoker how things are, and an emergency safety valve G is also fitted. The safety of and maximum pressure reached in the system are arranged by design. The heat dissipating surface is so proportioned with reference to the boiler heat intake surface that the former will get rid of heat at high temperature faster than the latter can absorb it.

The temperature control is somewhat indefinite, the low-sited pressure gauge is liable to be out of action, and the fireman—or more usually boy—has little to guide him except the plaudits or imprecations of the stove attendant. In spite of its crudity the system works well.

513. **STEAM BOILER SYSTEMS.** The Perkins system has been largely superseded by the use of an ordinary steam boiler. This, with its pressure gauge and safety valve, permits exact temperature to be obtained and held.

The search for smaller and cheaper power station boilers has produced certain long-tube boilers with forced circulation, such as the La Mont. These boilers have been hailed as the ideal for use with pressure hot water, because they already have circulating pumps. This is really a fallacy. If the circulating pump was the right size for the boiler alone, it cannot possibly do the very arduous duty of whizzing the water round the heating system. With a straight boiler, only the external circulating pump is needed, the feed pump is of course saved.

The real advantage of the La Mont type of boiler seems to be that in a pressure hot water system with a battery of boilers it is not necessary to have a drum to each boiler. One drum, or at the most two, will handle the water circulation from all the boilers on the range.

514. **WATER LOSS FROM P.H.W. SYSTEMS.** Theoretically there should be no water loss from a pressure hot water system. Consequently feed pumps, feed water treatment and all their attendant troubles should disappear.

The author, however, has heard of a pressure hot water system where quite a large feed pump, together with the spare pump, are hard put to it to keep the system full. The loss is partly due to men drawing off hot water for tea-making and other illegitimate pursuits. If the pressure is high the flash will be great and much extra water will be lost as flash. This may mean that for every 3 gallons drawn off 4 gallons must be replaced.

Draining points must be provided for emptying sections for maintenance. But these drains should be padlocked to prevent abuse. If maintenance, draining or water pilfering are heavy, many of the advantages of pressure hot water disappear.

515. FEED WATER FOR P.H.W. SYSTEMS. It is of great importance that the water put into a p.h.w. system be as good as possible. Any sludge thrown down in the boiler may be pumped round with the water and deposited on the heating surface, thus defeating the object of p.h.w. which is to improve heat transfer. Apart from coating heating surfaces, such boiler sludge or original scale and rust has been known to cause blockages in small pipes attached to such things as press platens. Blowdown is normally not admissable or necessary.

The water should preferably be distilled water because softened water can cause trouble at valve spindle glands. Any leakage causes flash, concentration and eventual deposition of salts in the gland, sometimes making the valve immovable.

516. ECONOMISERS AND P.H.W. SYSTEMS. As the water returns to the boiler in a p.h.w. system only some 20–80° F. below outlet temperature there may not be any scope for an orthodox economiser in a high temperature p.h.w. system. This handicap suffered by the p.h.w. boiler can generally be overcome by using an economiser for some other heating purpose—see Chapter 23—for example the economiser might supply heat to a lower temperature hot water system. The large water velocity in a P.H.W. economiser may offset the apparent disability of small temperature difference, and an economiser may sometimes be used in an orthodox manner in a P.H.W. system.

517. COMPARISON OF STRAIGHT STEAM, STEAM CIRCULATION AND PRESSURE HOT WATER. For all-round general heating purposes for space or process heating, a straightforward steam + condensate system is supreme. It has the overwhelming advantage that steam gives up its heat at absolutely constant temperature, and that the temperature can be adjusted instantly and accurately. Under most conditions steam can give up its heat much more rapidly than water.

By introducing some complications it is possible to make pressure hot water hold exact temperatures. The hot water can raise steam locally in completely closed vessels whose steam pressure can control a bye-pass in the pressure hot water circuit. If there is a demand for many exact temperature plants in a factory, it would probably be much better to go to a straight steam system. But if most of the factory is suitable for pressure hot water and only a few plants need exact temperature, the local raising of steam may be convenient.

For difficult heating applications where heat transfer is very slow with consequent negligible steam flow, steam circulation may show a great improvement, especially if the low heat transfer is due to a collection of air. Steam circulation is somewhat vague. Its effectiveness cannot be accurately foretold, as there is no easy method of measuring or forecasting the rate of circulation.

For difficult heating problems, especially where there are not large heating space volumes, pressure hot water may give much better results than steam

circulation. It cannot be satisfactorily applied where no temperature drop is permissible, nor to heating surfaces where the volume is large with reference to the heat transfer. Pressure hot water greatly evens out peaks because it can never supply a sudden demand as quickly as steam, and the large body of circulating water represents a big reservoir of heat—pressure hot water contains from 50 to 150 times the heat contained in an equal volume of steam. Boiler efficiencies can be better maintained with pressure hot water than with any kind of steam system. There should be no loss of water—except for minute leaks—from the pressure hot water system, consequently there should be no water softening problems and no boiler scaling troubles. The power needed by the boiler feed pump should be saved, but a good deal of power, dependent on the resistance of the circuit, is used for circulation. The circulating pump power may become enormous if the temperature drop must be kept very low and if the resistance is high. The boiler pressure must be very much higher than the corresponding steam pressure.

There is one type of factory that may find pressure hot water very suitable, namely the widely-spread factory of single storey buildings containing many small heat users. In such a factory the collection and return of condensate often does not pay, consequently the condensate and its flash are lost. All this loss would be avoided with pressure hot water.

The author believes there may be a great future for pressure hot water. It is probably the solution to district heating. It may displace steam in many awkward heating jobs such as oil tanks and press platens. A laundry for a Midland town has been designed using pressure hot water for all processes (this rather seems to be overdoing it).

* * *

CHAPTER 19

AUTOMATIC CONTROLS

I am the very slave of circumstance
And impulse.

BYRON. *Sardanapalus.* 1821.

BYRON unwittingly here summarised the characteristics of automatic controls. C. E. G. Simmons has stated the matter more precisely : " An automatic control is tireless ; it is also brainless. It cannot anticipate or profit by experience ". Here we have the fundamental clues to the success or failure of automatic controls.

518. LIMITATIONS OF AUTOMATIC CONTROL. An automatic control can only perform a mechanical operation which *our* experience has proved desirable and profitable. Automatic controls are so often enthusiastically installed as hopeful panaceas and disappointment too often follows. The controller does what lies in its mechanical power, but it was expected to think, which it cannot do. *We* have to do the thinking first, profiting by *our* experience. We must then limit the task of the controller to operations that are within its capacity.

It is of supreme importance that we give each part of the controller the simplest possible task. We can then string a number of controls together and get something which may look magical to the uninitiated, but which, when broken down into its constituents, is quite elementary. An intelligent man can control a complicated operation, using his wits and experience to meet exceptional conditions. Before an automatic controller can meet exceptional occurrences we must use our imagination and experience to enable us to embody in the control the mechanical qualities that will enable the controller to do our will.

If we visualise all the conditions we wish the controller to meet, and all the tasks we wish it to perform, and if we then simplify these tasks with all possible zeal, we can make an automatic controller that will carry out our wishes much better and more consistently than any man, however skilled, because the automatic control is tireless ; it does not get bored ; it is not interested in the result of the 2.30.

519. USEFULNESS OF AUTOMATIC CONTROL. Automatic controls are great steam and heat savers. They are not necessarily applied to the control of steam ; often they control the flow or condition of the process material. The elimination of peaks is one of the most effective heat savers, and automatic controls, correctly applied, are of the greatest use in smoothing out the peaks and valleys.

Automatic controls do not necessarily save heat. It depends upon whether they are designed to do something that will save heat. For example an automatic boiler control, fitted to a range of boilers subject to very fluctuating loads, will keep the pressure far steadier than any fireman ; but it will not necessarily save coal. In fact, one might almost say that it is bound to waste

coal, on a stoker-fired boiler, because the function of the controller is to keep the pressure steady at all costs, and these costs will often include incorrect combustion. But the steady pressure may be a great steam saver and on balance the automatic control may give an overall saving.

Automatic controls do not necessarily save labour. The automatic boiler control saves no labour. The fireman must still be there, the controller is only a far better tool than the fireman's unaided hands and eyes. If there are only a few automatic controls in a works it may even be that the automatic controls will increase labour, because automatic controls must be examined very frequently and given little periodic adjustments. There are of course many applications where an automatic control saves labour and coal and material. The whole thing boils down to a matter of cost. Will the control so improve things, be it pressure, density, temperature, flow, weight, time-cycle, etc., that the cost of upkeep of the controller and the interest on its first cost will be more than repaid by the improved results ?

There are some elementary automatic controls, which are found in every factory however small, and they are accepted as matters of course without their status as automatic controls being appreciated ; for example safety valves and tank float valves. If these humble controls are properly understood we are half way to understanding the most complicated controls and the difficulties surrounding their proper application.

The description of automatic controls could take the form of definitions of terms and functions, but this would be pretty hard going and would probably not be read. In this chapter this will not be done. Automatic controls of various kinds will be described and the terms and functions will crop up and be dealt with as we go along. We shall start with a detailed discussion of the most common types of simple automatic controller and end up with a description of something much more complex. Many of the controls to be described are at work in the author's factory, so that the descriptions are not just copied out of catalogues.

Much very useful information can be obtained from the makers' catalogues which contain many beautiful diagrams giving varied applications of all kinds of controls. Automatic controls have reached a very high standard, but the makers tend to imply, most naturally, that the installation of their particular control will solve all problems. We have got to make the controller's job easy and give it a task that is within its capacity ; then almost any of the well-known makes will work excellently.

In dealing with automatic controls we must use technical jargon. The names are not universally agreed and the author has had to select those words that seem to him most appropriate.

520. TANKS AND FLOAT VALVES. " What is a tank for ? " There are many answers ; here are some of them.

Engineer who installed the tank : " To permit wide fluctuations in draw-off to be accommodated without transferring them to the inflow."

Production manager : " To increase production."

Process labourer : " I gotter empty the bleeder."

Now the engineer is the only one whose answer gets full marks ; but has the engineer arranged his automatic control—to wit a ball float valve—so that his lofty specification can be complied with ?

In most factories tanks have a nasty way of being either full or empty. Their control arrangements are such that this is bound to occur unless man intervenes. If the tank is always full or always empty it might just as well not be there. The true function of a tank is to iron out fluctuations. The extent of the fluctuations that are to be smoothed out must be estimated or measured, then the size of the tank is at once patent. There can be no argument. The volume of the largest fluctuation that it is desired to smooth out is the tank size exactly. The fluctuations may be the fill of a bath or a month's water supply for London. Having got the right tank size, it is no use installing it in such a way that it is always full or always empty.

Many tanks are controlled by float valves. The float valve is worthy of detailed study. Consider the full-lined float F in Fig. 316 and assume that the plant conditions are such that the maximum outflow can be met by the maximum inflow. For every outflow the liquid level will drop to such a height that the inflow corresponds to the outflow. This means that the only part of the tank that is used is A C ; C D is useless, it is always full. A C is not really useful as tank, it is simply a method of opening the supply valve. The full line in Fig. 317 shows the relation between tank level and inflow.

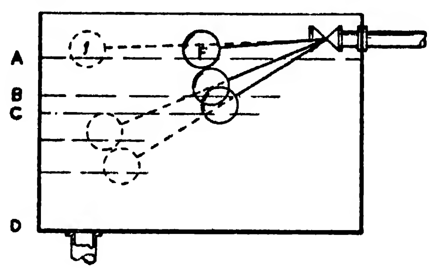

FIG. 316. PLAIN BALL FLOAT VALVE

There would seem to be only four reasons for fitting a ball float tank : to smooth out fluctuating draw-offs ; to smooth out fluctuating inflow ; to break the pressure head ; or to act as an emergency reservoir, for example, against fire (but this is not the kind of tank we are considering). An arrangement such as is shown by the full line in Fig. 316 will partly smooth out fluctuating inflows and is the imperfect solution to this problem only. If it is to smooth out intermittent or fluctuating draw-off we want to use the whole tank without transferring the intermittent demand to the inflow any more than is absolutely necessary.

We can lengthen the float lever and increase the RANGE of the control. We shall use much more of the tank and make the tank level/inflow characteristics follow the broken line *f* in Fig. 317. The lengthened float lever is shown broken-lined at *f* in Fig. 316. This arrangement does not iron out the fluctuations. It smoothes them out a bit, but it transfers the fluctuating draw-offs to the inflow and imposes a LAG.

It might be thought that by making the tank very much larger, a greater tank capacity becomes effective. This, however, is not so. If the tank is made very large the message to the float and valve will be delayed, a long lag will be imposed, and although short sharp peaks may be smoothed out, long peaks or valleys may be aggravated.

The ideal relation would probably be such that for nearly the whole of the range of tank levels the inflow rate would remain constant at the average flow. When the level gets dangerously low the inflow valve should open quickly and when the tank gets nearly full the valve should close quickly. Fig. 318 shows the characteristic level/inflow relation that these requirements call for.

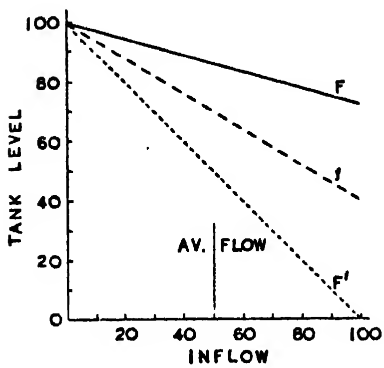

FIG. 317. FLOAT LEVEL/FLOW CHARACTERISTIC OF PLAIN FLOAT VALVE

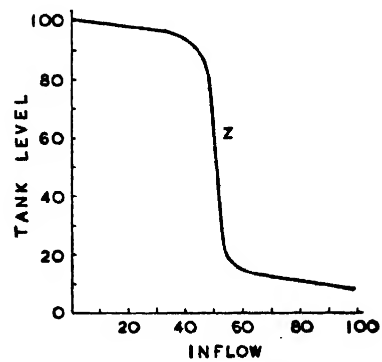

FIG. 318. DESIRABLE LEVEL/FLOW CHARACTERISTIC

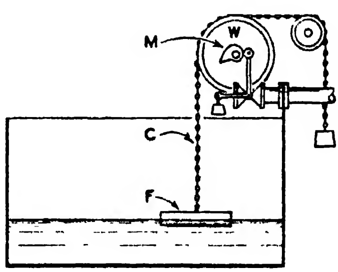

FIG. 319. FLOAT AND CAM CONTROL GIVES ANY DESIRED LEVEL/FLOW CHARACTERISTIC

There is no use fitting a ball float valve like that in Fig. 316 and expecting it to give a characteristic like that in Fig. 318. Fig. 319 shows one way of making a float give any desired level/inflow relation that we might want. The float F is allowed to travel the full depth of the tank. It is attached to a

chain C which passes over the wheel W. The wheel rotates the cam M which actuates the valve lever. By suitably cutting the cam any desired flow/level characteristic can be secured.

The effect of this float and cam control is to enable the tank to function as a tank about the average flow. But if the average flow changes, the tank will cease to function as a tank and will soon be either full or empty. By making the control more complicated we can make the control adjust itself to suit a changing average flow. This may be very necessary. The average flow in the morning may be double that in the afternoon ; the average flow on Friday may be half that on Monday. Such high-falutin' controllers are described later in Sections 551 to 556 after some of the other qualities of controllers have been examined.

521. DISPLACERS. Fig. 320 shows a float substitute which has many advantages. Floats are by no means satisfactory. The message delivered by a float is one of position only, the weight of the float is constant. Floats can puncture and waterlog. A float and chain such as are shown in Fig. 319 can jam or jerk. In Fig. 320 the float is replaced by the displacer D. The weight of the displacer is taken by the spring S. As the level of the liquid in the tank falls, more and more of the weight of the displacer is taken by the spring. The displacer can clearly deliver two messages ; either one of position or of weight. The force or weight message in Fig. 320 will be a straight line relation like that shown in Fig. 317 at F′. If the position message of the displacer is used we can connect the displacer to the valve directly and again we shall get a characteristic like F′ in Fig. 317.

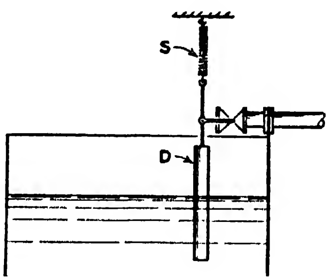

FIG. 320. DISPLACER TANK LEVEL CONTROL

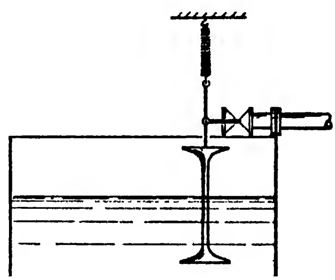
FIG. 321. SHAPED DISPLACER GIVES LEVEL/FLOW CHARACTERISTIC OF ANY DESIRED FORM

But we can make our displacer any shape we like. If we shape it like that in Fig. 321 we can get a characteristic which exactly follows the curve shown in Fig. 318.

We have done our thinking for the control, so that the displacer can blindly and tirelessly carry out our wishes. But, such a shaped displacer will only make the tank work as a tank if the average flow remains constant.

If the tank is to smooth out fluctuating inflows for a constant outflow or if it is to act as an emergency reservoir, a quick-acting valve, operated by float or short displacer, will probably be satisfactory.

In the W.C. flushing tank the float valve is not acting as a controller, but simply as an automatic refilling valve.

522. IMPULSE MODIFIER, DETECTOR AND REGULATOR. In all automatic controls, however complicated or however simple, there must be an IMPULSE which delivers the message that the conditions that are to be controlled have changed. This impulse must be received and possibly MODIFIED by a DETECTOR. The detected and modified message must then be translated into action by the REGULATOR. It may be that these functions are not cut and dried and carried out by separate parts of the device. In the plain float valve the impulse, detector and regulator are all carried out by the same thing—the float and its lever. In the displacer operated valve with a parallel displacer the impulse is produced by the displacer, the spring acts as detector and, by extending, allows the displacer to act as regulator. In a shaped displacer the displacer gives the impulse which is suitably and automatically modified by the shape of the displacer.

523. RANGE. A control such as this tank level control works through a RANGE. The range is the whole level of the tank. Suppose that for some reason it was essential to keep the tank as full as possible. This might be necessary where the function of the tank was to break the pressure and to apply a constant lower pressure head. In order to reduce the range of the float movement, the float would have to be greatly increased in size and its movement geared up by leverage to the valve. But, however big the float, there would always be some range. To limit the range to negligible proportions would call for such a large float as to be impracticable in many cases. There are ways of getting over the difficulties which will be apparent later.

One of the essential characteristics of any direct float or displacer control is that a particular float or displacer position corresponds with a particular flow rate. So that such a tank is a flow meter on a grand scale.

Clearly the ideal tank controller would be such as could permit the inflow to be steady, while the outflow was just what was drawn off, or vice versa. Over the bulk of the tank the one should not influence the other. If we secure such control our tank would be fulfilling its true functions. It would always have to try to reach an equilibrium at the temporary average rate of flow and it would have to ignore impulse messages when the level was high and falling or when the level was low and rising. We shall end this chapter with descriptions of two controllers which will fulfil these difficult conditions. They cannot be met by the simple float valve, or for that matter, any direct acting control.

* * * *

The float valve is full of interest and is seldom given the consideration it merits. It is too ordinary, too much like the waters of Jordan.

524. SAFETY VALVES. Fig. 322 is intended to represent the essential parts of an ordinary dead weight safety valve. The area of the valve when seated, Fig. 322a, is 1 sq. in. The weight is 100 lb. The valve should lift when the pressure exceeds 100 psi. Now when it lifts a minute amount, Fig. 322b the pressure on the ring B near the seating is less than 100 psi because at the

edge of the valve the steam has a free escape to atmosphere and the lifting force will decrease. The valve will therefore take up a position where the full pressure on the inner disc A plus the reduced pressure on the ring B is just sufficient to balance the weight, Fig. 322b. As the pressure rises the valve will lift a little more and the ring B will get larger, Fig. 322c. So that, although the load on the valve is constant, in order to get a reasonable outflow the pressure has to rise considerably above the set pressure of 100 psi. Such valves must always have a considerable range, which is just what we do not want a safety valve to have.

If the valve is spring loaded instead of loaded with a dead weight, the range is aggravated because, as the valve rises, the spring pressure increases, which calls for a greater lifting force.

Valves of this type almost always breathe. In order to give a safe relief to the boiler and give a sufficient flow at not too high a pressure, they generally have to be set so that they are breathing at normal working pressure. They are very wasteful for this reason and because, due to the constant breathing, the seat scores.

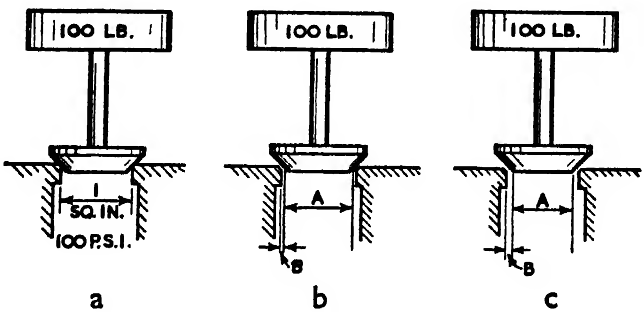

FIG. 322. ORDINARY DEADWEIGHT SAFETY VALVE

525. POP VALVES. Fig. 323 shows a safety valve with a modified seat. The actual valve seating is the usual inner conical portion and the opening pressure is exerted on the disc area A. There is a V shaped extension to the seat which has a small clearance between the fixed and moving parts. When there is sufficient force to lift the valve, the steam can only escape with difficulty and exerts a force on the V shaped ring B. This extra lifting force is added to the normal lifting force on the area A, so that the valve lifts quickly and high. As soon as the valve starts to lift its balance of forces is upset and it becomes unstable. It can only reach stability by opening sufficiently wide to reduce the lifting force on the ring B to restore a balance between the lifting force and the valve load. The valve must lift high if it lifts at all. The act of lifting increases the lifting force, whereas in the plain safety valve the act of lifting decreases the lifting force. The valve opens with a definite pop and closes bang. The valve operates over no range between shut and open, but once it is open, there must be a small drop in pressure before it can shut again.

Apart from the fact that the pop valve is either definitely shut or nearly full open and consequently the valve seat suffers little from wear, it has excellent psychological properties. The opening of the valve makes itself most blatantly noticeable. Everyone says " Blimey ! we're wasting steam." A plain safety valve whispers insidiously and its message is taken to mean " We've got a good head of steam ". The replacement of plain safety valves by pop valves is often a most fruitful way of saving steam. Apart from this saving, the boiler pressure can usually be raised by 5 or 10 psi. Where a plain safety valve may have a range of 15 psi the pop valve will probably only have a range of 2 to 5 psi. This means that with the same maximum pressure and maximum safety valve opening the boiler pressure can be 5 to 10 psi higher.

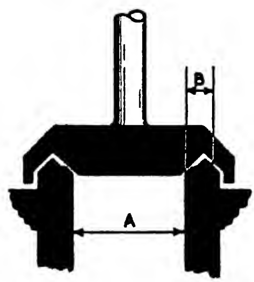

FIG. 323. THE PRINCIPLE OF THE POP OR QUICK LIFT SAFETY VALVE

There are many types of pop valve. Their makers often do not call them pop valves but by various names such as " quick lift ", " full bore ", etc.

The pop valve is an ON-OFF controller.

526. ON-OFF CONTROLS. It might be thought that any kind of on-off control would result in bad fluctuations, but this is often by no means the case. If the controller is such that the range can be very small an on-off control can operate very satisfactorily indeed. The small range causes the on and off periods to be very short and although the controlled result must be a series of ripples their amplitude is in many cases so small as to be of no practical account.

Thermostats often, in fact generally, work on the on-off principle and can often hold the temperature within $\pm 1°$ F. One of the most common and satisfactory on-off controls is on the voltage regulators of dynamos and generators. The impulse is a magnetic field which is proportional to the voltage. The magnet attracts an armature whose contacts short-circuit a resistance in the exciting circuit for a very short time. The contacts usually work at rates between 4 and 1 per second but the on-off action is entirely smoothed out by the sluggish response in the excitation due to magnetic inertia.

527. COMPENSATION. In Fig. 131, Section 292, we have a trap with a discharge capacity of 50,000 lb./hr. or nearly 100 gallons a minute at 250 psi.

Were such a trap direct acting the float would have to be enormous. So the trap is indirect and works through an oil operated relay. The float is quite small and simply opens or closes the oil leak C.

When the condensate rises in the trap body it lifts the float. Some small time interval elapses before the oil pressure rises sufficiently to open the main discharge valve. Meantime the condensate continues to rise. The valve then discharges full bore and the same lag occurs before the discharge valve closes. This HUNTING is so pronounced that this float trap turns itself into an intermittent discharge trap.

Now look at Fig. 130 in Section 292. When the water rises in the bottle B its increasing weight gradually overcomes the tension of spring D till the bottle drops and closes the oil leak E. The regulator valve immediately starts to open and at once the spring tension is increased due to the connection between the valve lever and the spring anchorage. This at once lifts the bottle slightly and allows a little leakage from E thus preventing further opening of the regulator. If the valve G has opened too little, the condensate will go on rising in the bottle B which will drop again, close the oil vent E, allow more oil into cylinder F, open the valve G still more and again tighten the spring D to prevent overshooting. If the valve G had opened too much the water level would drop in bottle B, the leak E would open sufficiently to allow the valve G to close slightly (it is trying to close due to the pressure in the trap tank and to the counterweight H). By finding the correct point K for attaching the wire and by choosing the correct spring tension the action can be completely COMPENSATED so that there will be no hunting and so that any rate of discharge that may be required will be obtained with a minimum range of level.

Compensation has its perfect application in a case such as this where the control is REACTIVE. A reactive control is one in which the *impulse* responds to the *effect* produced by the regulator. The impulse is water level. The regulator controls water level, so that the regulation reacts on the impulse.

Compensation can take all kinds of forms, but the essence of all compensators is that the detector runs away from the impulse. When the impulse says " Open ", the regulator opens but in opening pulls the detector away from the impulse and says " I won't open any more until you give me another impulse." Similarly in closing, the impulse tries to open the leak E but the effect of the compensator is to close leak E thus making the controller call for continued impulses before making further regulations. All compensators are largely matters of trial and error. It is difficult to foretell exactly the amount of compensation that a new application calls for.

The effect of under-compensation is to permit hunting and to encourage intermittent action.

The effect of over-compensation is to increase range.

With perfect compensation the range will be a minimum and the control will be as nearly continuous as possible. The plain lever float valve can be looked upon as being completely compensated. There is no lag in response to the impulse ; it is completely and instantly reactive ; the actual increase of water flow caused by the regulator reduces the impulse. Unless a tank can start

surging waves, the float valve will not hunt, but if the tank is such that surging waves can start, the float valve can hunt in an alarming way. Tank baffles will usually put this right.

There are some applications of control to which compensation cannot be applied. These are the systems that are NON-REACTIVE and some other types of indirect control. In such cases we have to use SUPPRESSION.

A non-reactive control is one in which the effect of regulator movement is not transmitted to the quality that sends out the impulse.

528. SUPPRESSION. The heat release in a boiler furnace is proportional to the fuel burnt, or to the oxygen consumed. We must control both, but we can control one after the other. If we control the coal the response will be sluggish. There will be a very big lag. If we control the air the response will be immediate, but unless the increase of air is backed up by an increased coal supply the incandescent coal on the fuel bed will quickly be burnt and the fire will get full of holes. A boiler controller should maintain the boiler pressure constant by controlling the air instantly and follow this up with control of the coal. The impulse must be, at least in part, a steam pressure impulse. The pressure impulse can be backed up or modified by a steam flow impulse. But whether the main impulse be pressure or flow of steam, the controller will act on air supply or coal supply or both. It is therefore impossible to make a boiler control reactive. If the pressure is low and dropping the fire may be large and increasing and it is no use providing compensation between the fire control and the controller. If the control were a simple control of fire impulsed by boiler pressure a drop in pressure might build up an ever increasingly great fire so long as the pressure were below normal. Although the pressure might be rising rapidly the great fire would go on being augmented all the time the pressure was low. When the pressure had been brought back to normal it would be impossible to damp down this fire in time, the pressure would overshoot and the boiler would blow at the safety valves. It is clear that in a boiler there must be a lag and there can be no compensation.

Now suppose we give the controller two impulses ; one, a simple pressure impulse—high, low, right ; and the other impulse, the rate of change of pressure and the direction in which it is changing. We can then make the controller bring the pressure back exactly to normal, we can SUPPRESS any overshooting and we can eliminate hunting and use a very small range.

The Kent master controller is a fine example of such a double impulse control which gives complete suppression of overshooting and hunting, yet is very easy to understand. The controller receives impulses of boiler pressure every 15 seconds and one of the impulses is the amount and direction of any change in boiler pressure during the previous 10 seconds. The detector translates these pressure impulses into secondary impulses which actuate the dampers or the speed of the induced draught fan. This gives an immediate heat release change and is followed up by separate subsidiary controllers on the forced draught and the stoker speed.

The master controller is shown diagrammatically in Fig. 324. The cylinder A is directly connected to the boiler and carries boiler pressure. In it is the beautifully fitting piston B. The piston B is held in the cylinder by the

weight C of such a size as to exert on A a pressure of just a minute fraction over boiler pressure. Shaft D is driven by the controller motor and rotates the piston B continuously so that it will not stick. The stationary weight C transmits its load to the piston through a ball thrust bearing. A very light spring G relieves the piston of a minute part of the weight. The result of the spring G and the fact that it is balanced in any position by the difference in pressure between that in cylinder A and the weight, gives the controller a range over a considerable movement for a very small pressure change, such as would be invisible on an ordinary pressure gauge. There is thus a fixed height for the

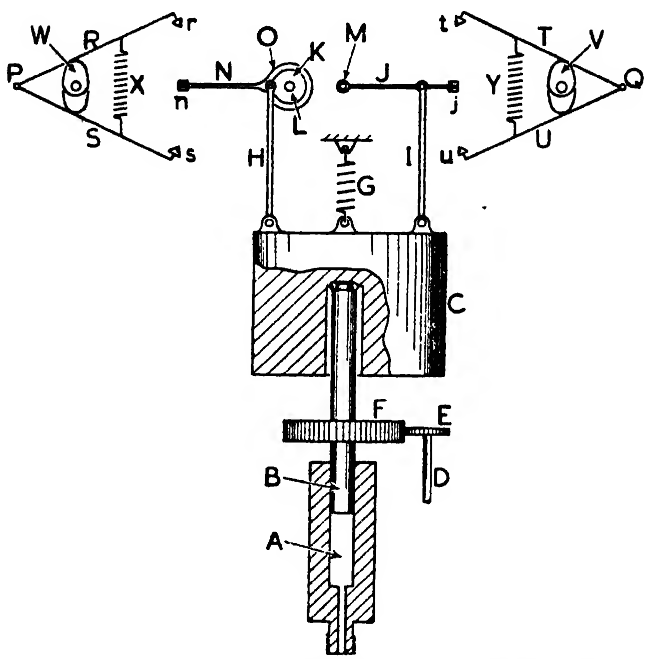

FIG. 324. THE KENT MASTER BOILER CONTROLLER

weight corresponding to any small variation in pressure either side of the working pressure. Connected to the weight by the links H and I are the contact arms J and N pivoted at L and M. Arm J is connected direct to the weight so that the angular position of J is a measure of the pressure in the cylinder A. The arm N is connected by a friction clutch O K to the link H and the leverage is such that the movement of N is four or five times as much as arm J for the same pressure change. Embracing the contacts *n* and *j* on the ends of contact arms N and J are the contact arms R and S and the contact arms T and U pivoted on P and Q. These contact arms R S T and U advance and recede scissor-fashion under the influence of cams W and V and springs X and Y. As arm N is connected to K by the friction clutch O K the arm N is centralised at every movement of contact arms R and S, so that at every cycle the contact *n* starts at the mid point. The cams are so shaped that they take 5 seconds to centralise and are then held in the open position, Fig. 324, for 10 seconds.

The result of this mechanism is that every 15 seconds the actual boiler pressure is measured by arms T and U and every 15 seconds the change in boiler pressure in direction and magnitude that has occurred in the past 10 seconds is measured by arms R and S.

Contacts *n* and *j* are connected together and to one supply point. Contacts *r* and *t* are connected together and lead to the "lower" relay. Contacts *s* and *u* are connected together and lead to the "raise" relay.

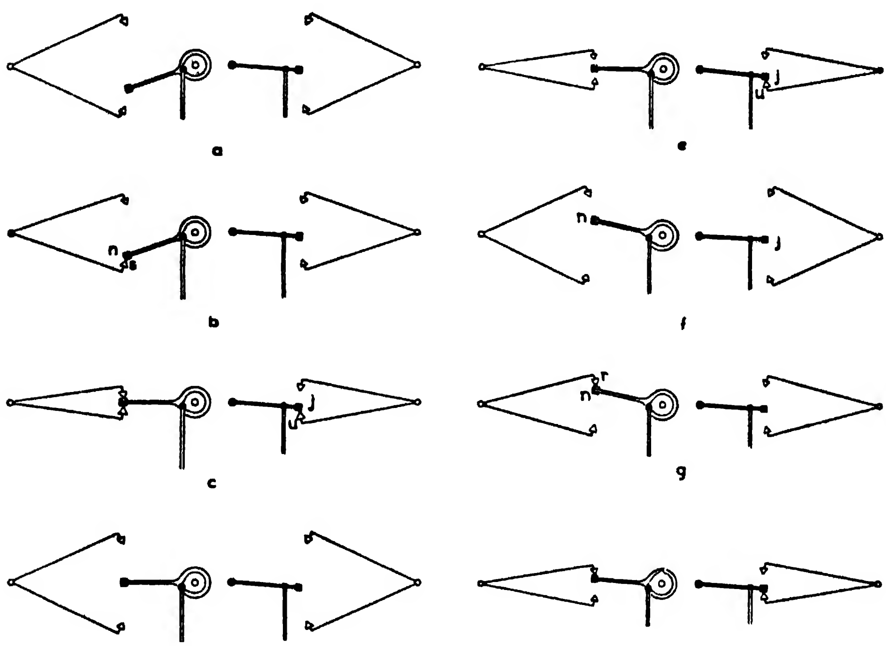

FIG. 325. THE ACTION OF THE KENT MASTER CONTROLLER

We will assume that the contacts have just opened and that the boiler pressure is correct. There is a sudden demand for steam so that 10 seconds later the weight has pulled the contacts into the position shown in Fig. 325a. The cams now allow the detecting contacts to close and in two seconds the "raise" contact *s* meets the "pressure rate of change" contact *n*, Fig. 325b. The "raise" controller receives a long impulse until the contact is centralised as in Fig. 325c. Just before centralisation the pressure measuring contacts *u* and *j* make and register a "raise" impulse, but as the contacts *n s* are also giving the raise impulse this makes no difference. The contact arms then open and wait 10 seconds.

If the pressure continues to drop, exactly the same sequence takes place except that contact *u* will meet contact *j* sooner, but if the correction applied by the controller is enough to check the fall in boiler pressure so that it remains constant at just below normal pressure the condition after the 10 seconds will

be as in Fig. 325d. The contacts then close and no " rate of change " impulse is given, but a short " raise " impulse is given by contacts *u* and *j* as in Fig. 325e.

If sufficient correction has now been given to make the pressure start rising fairly rapidly the situation will be as in Fig. 325f at the end of the next 15-second period. The boiler pressure contact *j* is still low, though not quite so low as before, but the rate of change contact *n* has risen. When the contacts approach, the first impulse is a " lower " impulse due to the meeting of contacts *n* and *r*, Fig. 325g. This is soon neutralised by the " raise " impulse when contact *u* meets contact *j*, Fig. 325h. Thus as soon as the conditions start to change the controller prevents them changing too much and performs the equivalent of compensation in a system where compensation is impossible.

This ingenious and very simple mechanism protects a system which is non-reactive, and which has an appreciable and variable lag, from hunting and SUPPRESSES all overshooting. It has other advantages in that its operation can be watched and its impulses can be made plain by means of pilot lamps in the raise and lower circuits. It can be used for all sorts of things other than boiler control—for anything in fact that is non-reactive and consequently calls for suppression.

It will be noticed that in the rate of change part of the detector contact *n* is centralised or RESET every 15 seconds. Any controller which measures rate of change and acts on it is called a RESET controller.

529. FOLLOW-UP OR SELF-CENTERING GEARS. These are hardly automatic controls. They are simply links in a system of control, but they are frequently used in automatic controllers and should be understood. Their original application was for operating a heavy mechanism by means of a light pilot control. Long before the word " Servo " was in common use, these steam servos were in constant use on shipboard for reversing gears and steering gears. In automatic controllers they are often present on a small instrument-like scale, as relays, servos or pilot valves. The word " Servo " is derived from " serf ", a slave.

A good example, which is more easily understood than a relay, is the reversing gear of a large reciprocating steam engine. Where valve gear is small and light it can be moved by direct manual means. Where it is rather larger it can be worked by hand through a worm reduction gear, but where it is very large, yet must be moved quickly, in the manoeuvring of a ship for example, it is necessary to have a light hand control which is exactly repeated by a heavy mechanical gear.

Fig. 326 shows diagrammatically a large marine engine reversing gear. A is the main engine valve stem. B is the Stephenson link which is rocked by the piston C in the steam cylinder D. The slide valve of cylinder D is shown at E ; it has no lap, and is connected to the end of the floating lever F. The floating lever F is connected by links to the hand lever G and the rocking lever J.

If the hand lever G is moved into the dotted position the floating lever F swings about pivot K and moves valve E forward admitting steam behind piston C. This moves the link motion over to the dotted position and lever J swings the floating lever F about pivot L until valve E closes. The main gear

FOLLOWS UP the hand gear until it takes up an exactly corresponding new position. If during running the link B moves out of position either way, the floating lever F will swing about point L, will open valve E a small amount and will restore the gear to the set position. This wandering, or "creeping" as it is often called, is well known on some marine engines. In this description it has been suggested that the main gear follows behind the hand gear. This of course is true, but the rate of follow is so quick as to be to all intents and purposes instantaneous.

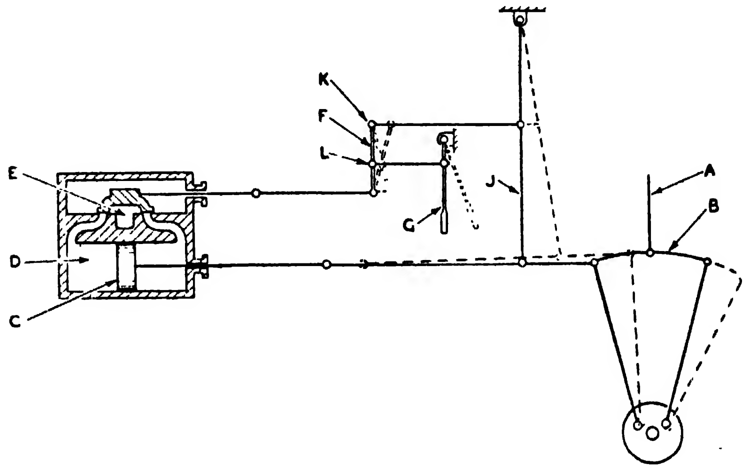

FIG. 326. STEAM REVERSING GEAR—FOLLOW-UP WITH QUICK RESPONSE

530. STEERING GEARS. The gear shown in Fig. 326 could work quite satisfactorily as a steam steering gear, if link B were replaced by a full sized helm and were G a miniature tiller. The mechanism would however have to be exceedingly large and cumbersome. The same principle is used in the ordinary steering engine, but, owing to the fact that the engine may have to make several or many strokes the valve gear must be more complex.

Fig. 327 shows diagrammatically the principle upon which most steering engines work. The engine must be two cylindered with cranks at 90° to ensure starting in either direction from any position. For simplicity only one cylinder A is shown in Fig. 327. The engine valve B is a hollow piston valve with little or no lap and with the eccentric set at 90° from the crank. Valve C is the control valve. When valve C is moved to the right steam is admitted to the engine steam chest through port F so that valve B becomes an outside admission valve. If the valve C is moved to the left steam is admitted to the valve chest through port G and valve B becomes an inside admission valve. An engine will of course reverse in direction if the inlet and exhaust valves are interchanged and this is what valve C does.

The control valve C is moved by turning its valve spindle which has a screwed portion H passing through the screwed boss of the worm wheel J. When the wheel on the bridge is turned, the valve spindle screws in or out of

worm wheel J and admits steam to the inside or outside of valve B. Steam is admitted to one or other end of cylinder A; the engine starts in the appropriate direction and turns the rudder gear L through the worm K. At the same time the worm M is turning the worm wheel J which screws the valve C back to the central point and shuts off the steam supply.

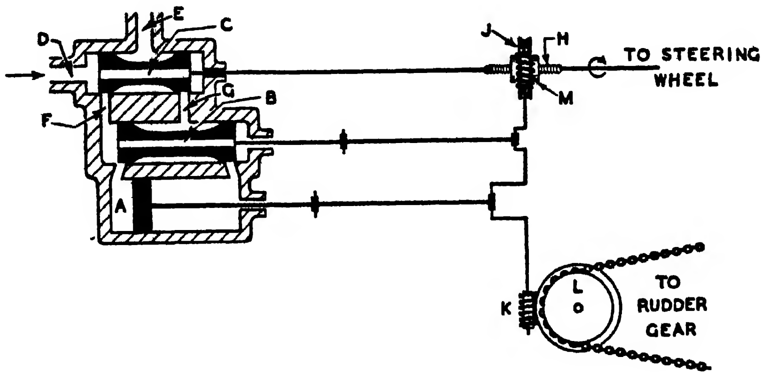

FIG. 327. STEAM STEERING GEAR—FOLLOW-UP WITH SLOW RESPONSE

However slowly the steering wheel is turned the engine will follow until it has restored valve C to the mid position. If the steering wheel is spun round hard a-port or hard a-starboard—sorry ! hard right or hard left—the engine will be on full throttle and will chase after the impulse, until it has caught up. The amount that the engine turns is exactly proportional to the amount the steering wheel has been turned. If by any chance the engine is running so fast that it overshoots, valve C will reverse and the engine will turn back to the right place. Overshooting is very unlikely because valve C is always being closed by the engine as fast as the man at the wheel is opening it. Like the reversing gear, this gear is inherently self-compensating.

This device is not an automatic control, but is sometimes a very useful device for inserting in the circuit of an automatic control system as a form of very powerful compensated relay. It is a beautiful, simple mechanism which has proved itself on thousands of ships over the last half century.

The reversing gear and the steering gear are examples of remote control coupled with great power magnification. In many operations great improvement in process operation would be secured by remote control of things like dampers and valves where power magnification is not needed. Hydraulic means, such as the Lockheed devices, can often provide a very neat solution to remote controls.

Gears of the kinds just described are to be found, on an instrument or model scale, on many automatic controllers. It will be seen that the follow-up movement of the main mechanism embodies a form of excellent compensation. Movement of the main gear resulting from the hand impulse is used to diminish

and eventually eliminate the impulse. There is however an important difference between the compensation in a follow-up gear and the compensation in a control such as that shown in Fig. 130. Fig. 130 shows a complete automatic control, where the impulse is produced automatically. The device shown in Fig. 326 is merely a relay on a grand scale. It is not reactive. The impulse originated in the skipper's brain, it does not spring automatically from the apparatus. The response of the ship neutralises the impulse in the skipper's brain. The real compensation should be between ship and skipper.

531. OPERATING MEDIUM. Almost all automatic controls must be indirectly operated if they are to be really satisfactory. Any complicated compensation, suppression, or the addition or subtraction of a number of impulses can be so much better done on an instrument scale than on the full scale. Controls are worked by hydraulic means, by water, oil or special liquids ; or pneumatically, by compressed air or steam ; or by electricity ; or by mechanical relays. All these systems have their particular virtues and particular applications.

Hydraulic. The great advantage of hydraulic operation is absolutely positive action. Water or oil are incompressible, so that if either be admitted behind a piston in no matter how small quantity the piston must move. As the water or oil cannot expand the piston cannot overshoot. If the liquid is oil, the whole mechanism is lubricated, but oil leaks are dirty and a return line to a sump or tank is needed. Water, however pure, tends sooner or later, to clog the small orifices of controllers. Sometimes the blockage is due to scale or solid dirt, more usually it is slime. Unless there is some good local reason for using water, oil is generally preferable. But water is cheap ; water leaks are less objectionable ; the small amount of water used can be wasted and there is no need to provide a return line.

Pneumatic. Steam is often very convenient in steam control gear. The principal disadvantage of steam is that it can and does condense and the condensate may be very troublesome. Steam is hot and somewhat messy compared to compressed air. Compressed air is clean, air leaks are free from objection, there is no need for a return line, but air is compressible and therefore the action may not be positive. On the other hand, the very fact that air is compressible enables some very tricky effects to be produced.

Electrical. Condensers and Wheatstone bridges can do all kinds of calculation for us. Electrical impulses can be made visible by pilot lamps. Electrical primary impulses can be incredibly small yet can be amplified to any extent by valves and relays. But electrical contacts are always troublesome or at least sources of weakness, and the final electric regulator drive is a high speed motor which calls for a reduction gear. Huge magnets or enormous solenoids are neither practical nor economic.

Mechanical. Mechanical relays, in the form of trips or clutches can be very convenient in many kinds of on-off control. There are several types of mechanical amplifier, one of which is described in Section 533. But for all-round usefulness mechanical controllers are not nearly so flexible or adaptable as hydraulic, pneumatic or electrical devices.

Taking all things into account, compressed air is possibly the best medium, but it may often be very desirable to mix it up with electricity. The outstanding advantage of electricity is that its action can be made visible whereas compressed air is hidden inside a cylinder. An electrical boiler control system does not necessarily work any better than a pneumatic system. But, in the electrical system, every impulse, in direction and duration, can be made visible on pilot lights. The fireman can see that the controller is doing its stuff and he can see just exactly what it is doing and can compare this with what he would do. The hydraulic or pneumatic systems work equally well but they hide their light under a cylinder and the fireman cannot see what is going on. Pressure gauges can be fitted at various points on pneumatic or hydraulic systems, but their message may be difficult to decipher and can never be so clear as that of a pilot lamp. As such pressure gauges are not essential, it is usually only a question of time before they are allowed to get out of action. It is important, especially in a boiler house, that the controller should remain the servant and not become the master and a good fireman will constantly use his brains and experience to help the brainless controller. The fireman cannot do this if he cannot see all the time exactly what the controller is doing, or has to wait until the boiler bursts before he realises that the controller is taking a day off.

532. PREVENTION OF STICKING. An impulse may call for a very small movement of the regulator. If the regulator has to move a valve whose gland or bearing is tight there is a risk that the valve will move in jerks. Similarly the detector may tend to move in jerks if it contains a piston. In Fig. 324 the method used by Kent for preventing sticking and jerky action was shown to be a continuous rotation of the piston. In some of the Swiss automatic controllers using oil as the medium, a small short-stroke valveless pump is fitted in the oil circuit so that all the hydraulic pistons are given a small continuous reciprocating movement. This is an excellent way of preventing sticking and ensuring instant response to the impulse. The resulting ripples on the regulator are so small and quick that no perceptible fluctuations can be seen in the regulated flow, but the continual movement does of course mean more rapid wear and more maintenance.

A more subtle method of preventing sticking is provided in the Foxboro controllers, which are so arranged as to be really on-off controllers operating so rapidly (it is impossible to detect the action by eye) as to be virtually continuous. If any part of the controller sticks, for example the pen on the recorder chart, the controller at once stays on or off for a fraction of a second, and the regulator gives a kick that unsticks the gear. This system, applied to the earliest possible link in the control chain effectively prevents any part of the controller mechanism sticking.

533. MECHANICAL RELAYS. Fig. 328 shows the Bush torque amplifier, or mechanical relay which can give back-lash-free amplification of 10,000 times, either as continuous rotation in either direction or for small angular displacements. It is used in the elaborate differential analyser which can solve complex equations that are intractable to ordinary methods.

The two drums, D*d* and D′*d*′, each of two diameters can rotate freely on the input and output shafts I and O. These drums are driven by chains continuously

and in opposite directions at a speed a little faster than the maximum speed of rotation of the input shaft. Round each of the four drums is wound a cord, like a Prony brake.

On the inside end of the input shaft is pinned an arm P, carrying a bar *p* to the ends of which are fastened two very light cords wound round the small diameter of the drums *d d'*, in opposite hands. The other ends of these cords are attached to a bar *q* carried by the double arm Q which can rotate freely on the output shaft O. At the opposite end of arm Q is a bar *q'* carrying stout cords wrapped opposite handedly round the large drums D D'. The other ends of these stout cords are attached to a bar *r* carried by the arm R which is pinned to the output shaft.

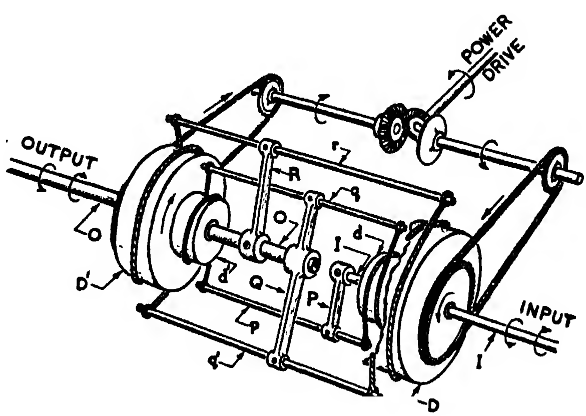

FIG. 328. MECHANICAL RELAY

When the input shaft rotates it will tighten one of the light cords which will grip the small drum and this will turn arm Q in the same direction as the input shaft with appreciable torque magnification. Arm Q will tighten the appropriate stout cord on the large drum. The stout cord will be carried round dragging the arm R and turning the output shaft with considerable force.

Provision, not shown in the diagram, must be made to adjust the tension of the cords. It does not matter, from the sensitivity point of view, about the braking effect of the cords because the power drive can be as powerful as is convenient. The only load on the input shaft is that needed to tauten the light cord, which can be made extremely delicate. The light cord only need be sufficiently strong to tauten the heavy cord.

The successful use of this device on an extremely accurate machine is a guarantee of its sensitivity and freedom from back-lash.

During the war many most ingenious mechanical amplifiers have been devised for aircraft controls. Some of these are mixtures of mechanical and electrical, or mechanical and hydraulic, but the device shown in Fig. 328 is a purely mechanical amplifier, relay or servo.

534. PRE-DETERMINED SEQUENCES. These are hardly automatic controls in the true sense in that they do not limit varying quantities or deal with an unprogrammed event, nor do they try to restore conditions to a given state. They merely carry out a programme, which may be variable, but must be preset at least in part. Notable and complex examples are the Jacquards and Dobbies on looms and the perforated rolls on piano players, or monotype machines. There are other types which can be useful in the steam using factory. Here is an example :—

535. FILTER PRESS PRESSURE CONTROLLER. Filtration in a process factory can, if badly done, be a great steam waster, or, if well done, a great steam saver. For example good filtration of a sugar solution enables the solution to contain only 30 per cent. of water, bad filtration may easily call for 35 per cent. of water to reduce the viscosity of the liquor to enable it to pass through the filter at the necessary rate. All this extra water must be evaporated by the use of steam.

30 per cent. of water on 8,000 tons of sugar a week is	3,430 tons of water.
35 per cent. of water on 8,000 tons of sugar a week is	4,310 tons of water.
Extra evaporation	880 tons of water.
or	110 tons of coal a week.

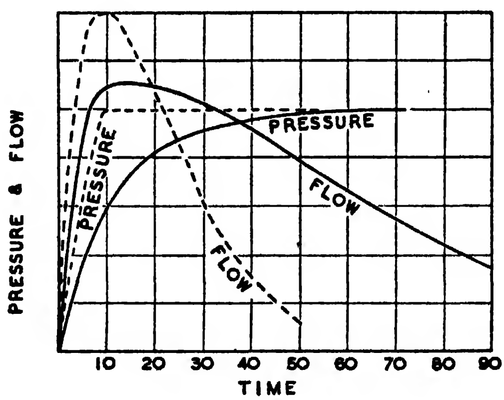

FIG. 329. PRESSURE/FLOW CHARACTERISTIC IN FILTER PRESS

It is important that the filter press be started up gradually and smoothly so as to build up an even porous cake. The pressure/time curve should be something like the full-line curve in Fig. 329 which also shows the approximate flow—also as a full line—that would be obtained. Too quick an application of pressure packs the cake hard on to the filter cloth and, though causing a greatly increased momentary flow, causes the flow to fall off rapidly. The broken lines

in Fig. 329 show the pressure/time and flow/time relations with a quick rise of pressure. In order to get the throughput, the viscosity and hence the density of the liquor must be reduced by the addition of water which must all be subsequently evaporated. The alternative method of reducing viscosity is to increase the temperature which causes the destruction of sugar and is even more undesirable than extra evaporation.

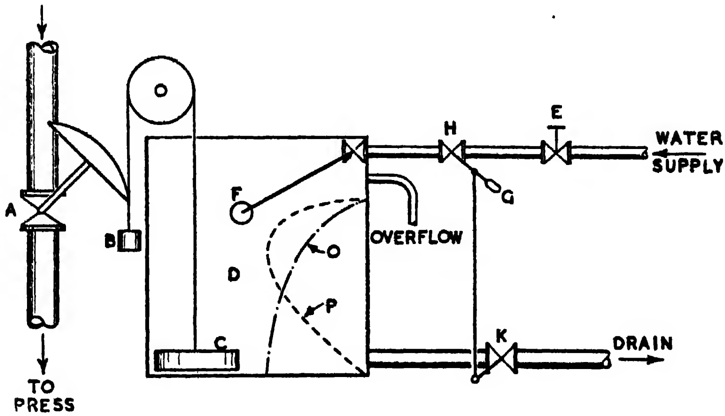

FIG. 330. FILTER PRESS PRESSURE CONTROLLER

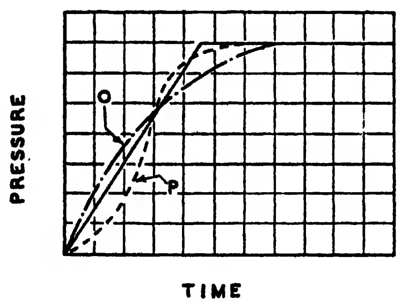

FIG. 331. PRESSURE/FLOW CHARACTERISTICS WITH PRESSURE CONTROLLER

The simple device shown in Fig. 330 gives a gradual, even, pre-determined but adjustable pressure application. A is the liquor feed valve to the filter press. This is shown in the shut position. The valve is operated by the weight B and the float C. The float is caused to rise, at any pre-determined speed, by a supply of water to the float tank D. This supply of water is set for rate of flow by the valve E. When the float rises to the top and the valve A is fully open, the float lifts lever F and shuts off the water supply. When the run of the filter

press is finished, the hand lever G is turned through 90°. This cuts off the water supply to the tank by means of cock H and opens the tank drain cock K. The float falls quickly but steadily and closes valve A without shock.

Any characteristic time/pressure curve can be achieved by putting a shaped filler into tank D, so as to alter its progressive cross section. Suppose the plain tank D gave a straight pressure/time curve like that shown as a full line in Fig. 331; by putting the filler O, shown chain-dotted in Fig. 330, into the control tank the pressure/time curve can be altered to that shown chain-dotted in Fig. 331. By fitting a filler shaped like the broken line P in Fig. 330 the pressure/time characteristic can be made to follow the broken curve P in Fig. 331.

To start the press all that has to be done is to move hand lever G back 90° and so close the tank drain K and open the control water supply cock H.

This is a crude but simple arrangement. It could have been replaced of course by a proper automatic controller in which a clock, a cam and a compensated pressure regulator could have opened the valve A.

The tank, control valve, piping, etc., and a recording pressure gauge cost £180, labour £113 in 1943.

Although the device gave the required pressure/time characteristic, it was a complete failure, apparently because the throttling disturbance at valve A broke up the precipitate floc and undid the good done by the gradual pressure rise.

* * *

A number of different true automatic controls will now be described. As far as possible the more simple controls will be dealt with first. It is difficult to arrange these descriptions in a really logical order, but it is hoped that the following sections will not be found too much of a hotchpot.

536. AUTOMATIC DENSITY CONTROLLER. Mother syrup after separation from sugar crystals is supersaturated and contains very fine crystals in suspension. These fine crystals must be dissolved and the syrup must be brought down to below saturation lest crystals be deposited in tanks and pipes causing eventual blockage and much labour for their removal. For subsequent reprocessing the density must almost always be reduced to at least 70 per cent. solids in solution. Whatever the density fixed upon, it must be adhered to as closely as possible because too high a density so increases viscosity as to hang up the next process, while too low a density calls for unnecessary subsequent evaporation. To do the density reduction manually is difficult. It necessitates taking temperature and hydrometer readings with consequent lag in control. If the man makes a mistake by letting the syrup go at too high a density he immediately sticks up the process and calls forth the wrath of the process foremen and managers. If he errs on the dilute side retribution is delayed until figures for steam, etc., have been worked out. He is therefore human and hopes that if his errors are on the dilute side he may get away with them and he generally errs to the extent of about 2 per cent. of unnecessary water. At this particular point in the process about 6,000 tons of sugar a week are dealt with in the author's factory.

30 per cent. of water on 5,000 tons of sugar a week is	2,570 tons of water.
32 per cent. of water on 5,000 tons of sugar a week is	2,820 tons of water.
Extra evaporation	250 tons of water.
or	30 tons coal per week.

The syrup density is automatically controlled in a very satisfactory manner by the device described below.

A column of water is hydrostatically balanced against a column of syrup. The balancing is done by means of compressed air bubbled through the columns. In order to prevent errors due to lag or errors in sampling the whole of the throughput is passed through the syrup column. Changes in syrup density cause changes in the relative air pressure needed to bubble the air through the liquid columns. These variations in relative air pressure are used to operate the valve admitting the diluting water.

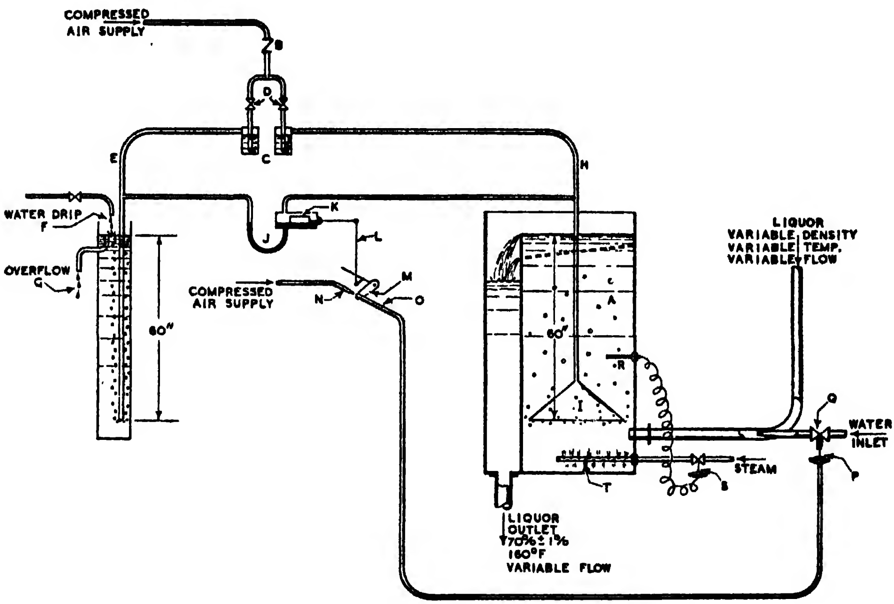

FIG. 332. AUTOMATIC DENSITY CONTROLLER

The syrup column takes the form of a long narrow weir tank A in Fig. 332. The object of this shape is to reduce the volume of the column as much as possible while giving the maximum length of crest for the weir. The weir is of such a length relative to the flow that the crest level does not vary by more than $\frac{3}{16}$in. between full flow and zero flow. This introduces an error due to column height in a 60 in. column of ·3 per cent. Air is supplied at low pressure through the reducing valve B to the two bubblers C. These bubblers are just small glass bottles half full of water through which the air passes so that the amount of air can be controlled by eye by means of valves D so that each column receives the same small quantity of air. The left hand pipe E leads to the bottom

of the water column which is kept at constant height by means of the water drip F and the overflow G. The right hand pipe H is connected to the air bell I in the syrup column. Equal amounts of air are passed through the two liquid columns. The air pressure needed to bubble air through the liquid columns of constant height is proportional to the density of the liquids. The two pipes E and H have connections to each arm of the mercury U-tube of a Kent flow meter. The pressure in pipe E is always the same. Any variation in pressure due to variations in density of the syrup in tank A cause changes in the position of the mercury in the U-tube. These U-tube changes cause the float K to rise or fall. If the syrup is too dense the float will fall. This will lift the link L which, due to the lever multiplication will lift the chopper blade M an appreciable amount for very small changes in the mercury level. The chopper M is interposed between the air jet N and the air pipe O. As the chopper M rises air blows across the gap into the air pipe O. This puts a pressure on the diaphragm P and opens the water valve Q diluting the syrup and reducing its density. As the density varies considerably with temperature and the syrup temperature fluctuates, the thermostat bulb R in the syrup column A operates valve S and admits steam to the perforated blower pipe T maintaining the temperature constant at 160° F.

The device can handle variations in density, flow and temperature and turn out syrup at a reasonably constant density with a constant temperature. There is no compensation so that the controller hunts continually and rapidly. The fluctuations are so small and so rapid that they are of no consequence. When first installed elaborate means were taken to try and ensure intimate mixing of the syrup and the diluting water, but this was found to be unnecessary and was dispensed with. Three of these controllers saved 25 tons of coal a week and a man on each shift, as well as giving more consistent process results.

537. SELECTIVE TEMPERATURE CONTROL WITH OVERRIDING FLOW CONTROL. In a sugar refinery the principal heat use is the evaporation of water during the crystallisation of sugar in the vacuum pans. The vapour is condensed by river water in jet condensers and the heat in the vapour is lost. For some 40 years in the author's factory much of this waste heat has been recovered in heating up all the incoming town water (for process and boiler feed) to about 125° F., by means of surface condensers inserted in the vapour pipes between the pans and the jet condensers.

As sensible heating by waste latent heat is one of the most fruitful sources of heat economy, the actual saving will be worked out :—

Water used per ton of sugar	400 gallons
Weekly sugar throughput	12,000 tons
Weekly water consumption	4,800,000 gallons
Heat added per gallon	700 Btu
Total weekly heat saved	3,360,000,000 Btu
Coal equivalent at 82 per cent. boiler efficiency and 95 per cent. plant efficiency	164 tons coal
a week or over £40,000 a year	

In 1939/40 it was decided that this water, before going to process or boiler feed, could do all the factory space heating, all the process air heating, all the

air conditioning heating and could also supply the social hot water to some 200 baths, showers, basins and sinks. The water is circulated rapidly round a circuit comprising the heating systems and the pan condensers, the process and boiler water being bled off after the condensers. (The system is shown in Fig. 418 and described in Section 761.)

The vacuum on each pan varies according to its process state between about 20 in. and 26 in., so that the vapour temperature varies between about 125° F. and 160° F. The water demand fluctuates somewhat violently due to certain necessary batch operations. The control therefore must allow whatever flow is called for to flow through the condensers but should ensure that the most water should pass through the hottest condenser. The arrangement is shown in Fig. 333.

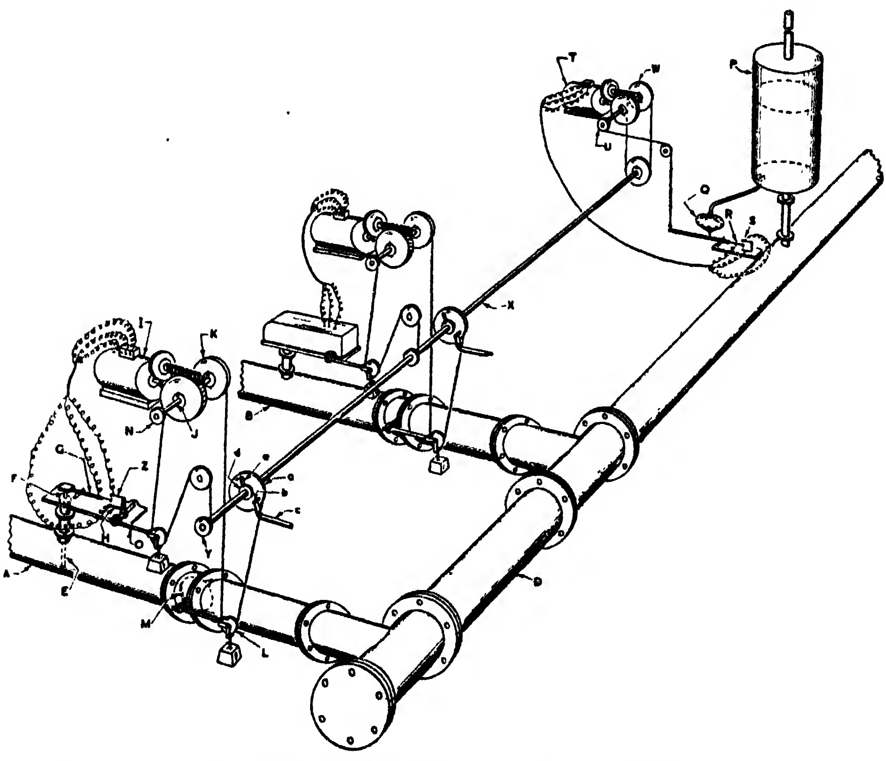

FIG. 333. SELECTIVE TEMPERATURE CONTROL WITH OVERRIDING FLOW CONTROL

Pipes A, B and C (C is not shown) lead from the three surface condensers into the manifold pipe D which leads to the heating circuit, the process and the boiler house. The water from A passes the thermostat bulb E. If the temperature of the A water rises the thermostat expansion rod F rocks the contact blade G and makes the contact H. This starts up the Hopkinson–Carlstadt split field motor I in such a direction that the double reduction gear shaft J rotates counter-clockwise. The wire attached to pulley K lifts the lever L and opens the butterfly valve M to allow more water to pass through the condenser as it gets hotter.

The wire attached to the small pulley N on shaft J pulls up lever O which draws the contact H away from the contact blade G and thus provides compensation.

This part of the mechanism therefore passes more water through any condenser that gets hotter and is provided with compensation to eliminate hunting and overshooting.

Tank P is connected to the main manifold pipe D and is at the topmost point of the system. When the valve M is opened and more water is allowed to pass through the branch pipe A, the level in tank P rises and exerts a greater pressure on the diaphragm Q. This rocks the contact blade R towards the contact S. This starts the split field motor T so that the pulley U turns clockwise thus rocking the contact blade R so that contact R is drawn away from contact S providing compensation. At the same time, pulley W is rotating clockwise. This turns the countershaft X clockwise and with it the pulley Y. Pulley Y pulls up lever O and draws contact H away from the contact blade G and perhaps brings contact Z into contact with blade G. This reverses the motor I and cuts down the water flow. But the flow reduction due to the rotation of shaft X is applied to all three condensers and immediately motor I starts the compensation will break contact GZ. The effect therefore is to cut all the flows but to maintain the larger flow through the hotter condenser, and to compensate this flow control.

We have individual thermostats trying to pass more water through their condensers the higher the condenser temperature. We have the control tank P adjusting the total flow so that it meets the demand.

Now suppose condensers B and C are both cold because for some reason the pans to which they are connected are temporarily off together. Condenser A will not provide sufficient water even if its thermostat opens valve M fully. The level in the tank P will fall, diaphragm Q and contact arm R will start the motor T so that it turns the shaft X in a counter-clockwise direction. This lowers the adjustment of all the thermostats. But the cold condensers B and C still do not respond and open their valves. The other end of the wire that is attached to pulley K is attached to pulley *a* which is free to rotate on shaft X but is normally prevented from turning by the stop *c*. Collar *d* is keyed to the shaft X and when shaft X has turned sufficiently to bring the thermostats into action, further turning causes pin *e* on collar *d* (and the corresponding pins on the other gears attached to the other condensers) to engage pin *b* and rotate pulley *a*. This opens the valve M, and the valves on the other condensers until a sufficient water flow has been produced. The water will of course not be so hot because some of it will have passed through cold condensers.

The amount of radiator heating is about 10,000 sq. ft., the air heated in conditioning plants and driers is about 175,000 cu. ft./min., and some 200 taps are fed with hot water. It is estimated that a steam saving equivalent to 35 tons of coal a week in winter has been secured by this system.

This controller was designed in the author's factory and made from standard Hopkinson components. The cost of the regulators, valves, thermostats, shafting, pipes, etc., was £381 in 1940. The labour cost of installation was £373, and there were some 200 hours of draughtsman's time.

* * *

One of the advantages of using an indirect control through an operating medium is that it enables all kinds of different impulses to operate the control. Some of these controls are very complicated, and, although really simple to follow if worked out step by step are rather daunting. We will make a gradual approach.

538. RUTHS ACCUMULATOR CONTROL. Section 444, Chapter 16, described the requirements for controlling a Ruths accumulator. Look at Fig. 334 which is a diagram with the controller shown very large, the regulator small and the accumulator tiny—for clarity. The valve A supplies the high pressure steam for the low pressure main direct or for charging the accumulator. The impulse *a* is the surplus impulse and must try to open the valve when the high pressure main is above say 148 psi. The impulse *b* is the reducing impulse and must open the valve when the accumulator pressure is below say 27 psi. The impulse *c* is the overriding safety impulse which prevents impulse *a* from operating if the accumulator is fully charged—that is when its pressure is up to 100 psi. Actually the Ruths controller is different from the controller shown in Fig. 334. The reason for describing the Ruths control by means of the imaginary controller shown in Fig. 334 is to introduce the subject of multiple impulses in an easy way.

The oil pump B draws oil from the sump tank C and pumps it into the pressure line D. The pressure line has an ever open connection to the top of the regulator cylinder E. The pressure pipe is also connected to two vents or leaks G and H which can be throttled or closed by the rockers K and L. When both vents are throttled or closed by the rockers the oil pressure drives the piston F down and opens the valve A by means of the wire M. Valve A can only be opened when both vents are throttled. If either vent is opened, the pressure is released and the valve A closes due to the weight N which pulls the piston F up and drives the oil out of the cylinder through the open vent. The oil runs back to the sump C through the drain O in the controller. When the valve is to remain stationary, the vents must be so throttled that they can pass the output of the oil pump and no more. Then no oil can enter or leave the cylinder so that the piston does not move.

All the time that the accumulator pressure is below 100 psi the impulse *c* acting on the bellows in chamber *c* is not strong enough to overcome the pressure of spring P. Vent G is therefore always closed until the accumulator pressure reaches 100 psi. As soon as the accumulator pressure reaches 100 psi the pressure in chamber *c* is enough to overcome the spring P and vent G is opened. All the time that the accumulator pressure is at 100 psi vent G will be open, the oil can escape from the regulator cylinder and the valve A will be shut, and nothing that the surplus and reducing impulses *a* and *b* can do will make it open. So long as the accumulator pressure is below 100 psi the vent G will be closed and the operation of valve A will be solely controlled by impulses *a* and *b*, whose goings on we will now study.

Vent G is closed and takes no part in affairs. Rocker L is acted on by two pairs of knife edges. Impulse *a* comes from the high pressure main and at, say, 148 psi the pressure in chamber *a* is sufficient to move the bellows against the pressure of spring Q. Rocker L will move down and close the vent H,

using the R spring knife edge as its fulcrum. When the high pressure drops below 148 psi the bellows will retire into chamber *a* until it comes up against the adjustable stop V. As the *a* bellows retires vent H opens, releases the oil pressure in the regulating cylinder and closes or tends to close valve A. If the *a* impulse (in reverse) is sufficient the valve A will be completely shut. The accumulator will then discharge until its pressure drops to, say, 27 psi when the spring R will overcome the 27 psi pressure in chamber *b* and the rocker L will rock, using the *a* bellows knife edge as its fulcrum and cause steam to blow over from the high pressure main regardless of the pressure in the H.P. main.

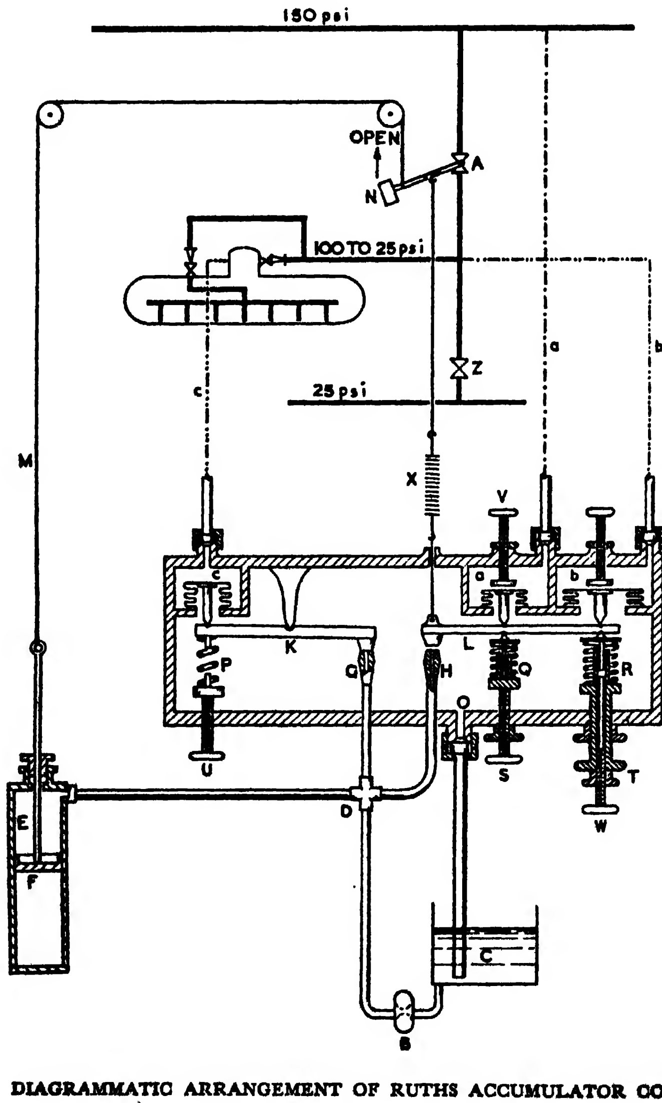

FIG. 334. DIAGRAMMATIC ARRANGEMENT OF RUTHS ACCUMULATOR CONTROLLER

During the normal action of the accumulator when it is varying between nearly full and nearly empty, the impulse *a* is the only one that is operative. When the pressure in chamber *b* is at or above 28 psi the bellows in chamber *b* is pushed downwards against spring R until the stem of the spring knife edge is brought to rest by the adjustable stop W which screws in or out of the R spring adjustment T. So that so long as the accumulator pressure is above 27 psi the impulse *b* is out of action, rocker L fulcrums on R and impulse *a* takes charge.

Attached to the end of rocker L is a wire with a spring X inserted in its length. This wire moves with the valve A and provides compensation for both impulses *a* and *b*. By choosing the right spring strength and extension, and by fitting the wire to the right point on the valve lever, any desired range can be obtained and all hunting can be eliminated. Impulse *c* needs no compensation. It is an on-off impulse and must have no range and therefore no compensation.

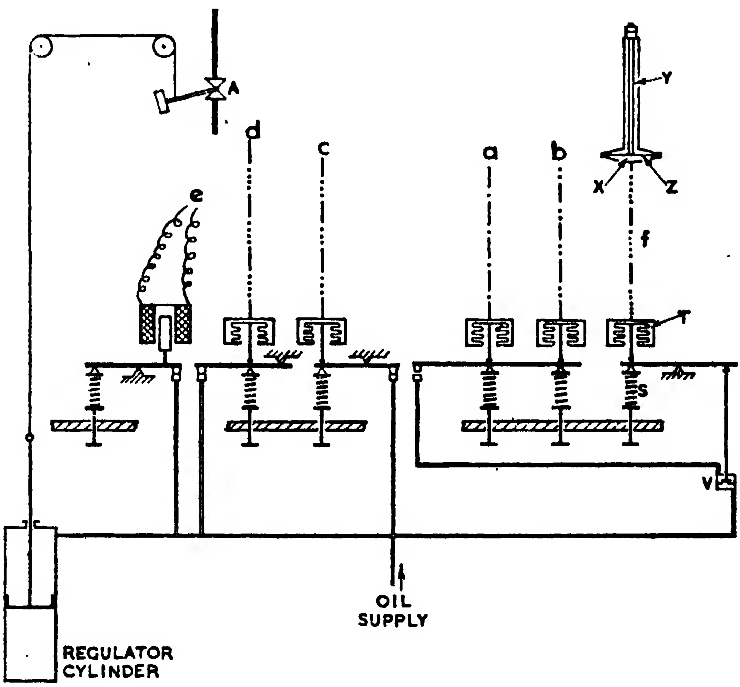

FIG. 335. MULTIPLE IMPULSES TO CONTROLLER

539. MULTIPLE IMPULSES. It is obvious that with a controller of this kind it would be possible to add all kinds of other impulses. For example we could add the three extra impulses shown in Fig. 335. Impulse *d* is so arranged that as soon as the high pressure main drops to 100 psi the valve A will be closed regardless of any frantic requests for steam from impulse *b*. Such an arrangement might be necessary to protect the high pressure line, which might be supplying an important engine, from sudden pressure drops. Impulse *e* is the

dynamo load and is detected by the solenoid shown and might be set so that whenever the electrical load reached 500 kW the valve A would shut so as to give the engine all possible steam.

It will be noticed that these two additional impulses close valve A. Any additional impulse can be very easily arranged to close the valve because all the impulse detector has to do is to open an additional oil vent. It is much more difficult to introduce an additional impulse that will open valve A. Multiple impulses can be made to open the valve very easily if the regulator is reversed, so that the oil pressure shuts the valve and the weight opens it. Such an arrangement is however so dangerous that it should never be considered. Oil pump failure, leaks, burst oil pipes would all cause the valve A to open fully. Automatic controls should always be designed to "fail safe".

It is possible to provide an impulse which will open the valve and override some or all of the other impulses, if this new overriding impulse operates a little oil valve in the oil pipe leading to some or all of the vents. In such an arrangement the impulse must be more powerful. The arrangement is shown in the case of impulse *f* in Fig. 335. The impulse *f* will override the impulses *a* and *b* in such a way as to open valve A even if impulses *a* or *b* call for closing. Suppose the principal process heat user is a most important temperature-holding process where loss of steam pressure would mean the spoiling of much material. Then impulse *f* will demand steam from the high pressure system even if the high pressure system had dropped seriously in pressure and although the accumulator were not empty. It is unlikely that such a condition could often occur, but it is possible, if the accumulator is of relatively small capacity and if its discharge rates are substantially below the average flow of steam from the high pressure to the low pressure system.

The temperature of the important heating process might be measured by the thermostat expansion rod Y which presses on the diaphragm Z which thus puts a pressure on the bellows chamber T. The bellows overcomes spring S and lifts the valve V thus supplying oil to the vent operated by impulses *a* and *b*. If the temperature drops, rod Y will contract and relieve the pressure on the bellows T, which will retire into its chamber and allow spring S to close the valve V. This puts both the *a* and *b* impulses out of action leaving only the emergency impulses *c*, *d* and *e* in operation. Unless an emergency electrical load or abnormally low high steam pressure is occurring the valve A will open and supply the low pressure main.

540. IMPULSE DETECTION AND AMPLIFICATION. The oil jets squirting from the vents on the controllers shown in Figs. 334 and 335 exert forces on the rockers and these forces must influence the sensitiveness and response of the controller. As the oil leak must be considerable and the oil pressure fairly high this disturbance will make the controller call for stronger impulses; or the bellows chambers must be large.

In order to permit a very weak impulse to be correctly interpreted by the controller it is desirable to limit the disturbing force introduced by the oil jet, and to reduce the jet itself to a very fine bore. This would mean that the response of the regulator would be very slow. So in nearly all controllers an amplifier is introduced. This is called a relay or servomotor.

In some controls efforts are made to reduce the interference by the operating medium of the impulse detector. In one case, the Ruths, an almost perfect solution has been found. It is, however, not absolutely necessary to eliminate interference, though elimination increases the sensitiveness of the controller to very weak impulses and this may be particularly important in some cases.

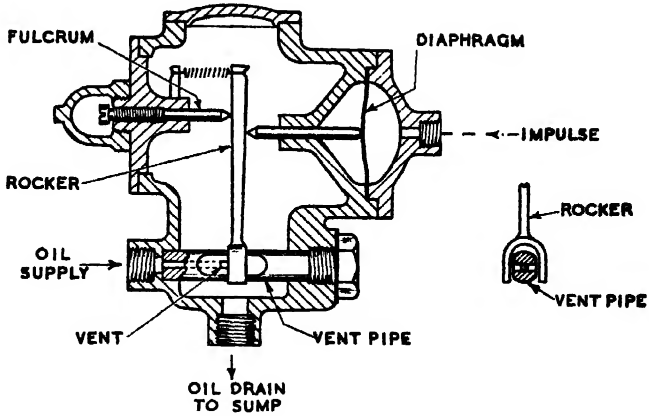

FIG. 336. RUTHS DETECTOR FOR SURPLUS IMPULSE

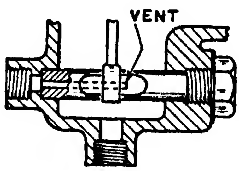

FIG. 337. RUTHS DETECTOR FOR REDUCING IMPULSE

541. IMPULSE DETECTION—RUTHS. The Ruths controller operates with oil as its medium, and Fig. 336 shows a simple Ruths single-impulse detector. The impulse pipe is connected to the diaphragm chamber and the impulse pressure acts on the diaphragm. The diaphragm movement is transferred through a short rod to the rocker which is pivoted on an adjustable fulcrum. The fulcrum point is only a very short distance—much less than is shown in Fig. 336—above the point at which the diaphragm acts. This permits a very light spring to oppose the impulse pressure, greatly multiplies the movement of the lower end of the rocker and makes the detector very sensitive. The spring is not adjustable but can be quickly changed by taking off the top cover. This lack of spring adjustment is possibly advantageous as the fewer knobs there are to tempt the amateur experimenter the better. The lower end of the rocker is forked and embraces the vent pipe whose sides have flats

cut on them. The vent is formed by a hole drilled across the vent pipe passing through its central hole. The forked end of the rocker does not touch the vent pipe, but is just a clearance fit. There is an equal jet pressure on each prong of the fork so that no jet pressure can be transmitted to the rest of the mechanism and cannot react on the detecting diaphragm. The pressure impulse in Fig. 336 closes the vent as the impulse pressure rises and the device must therefore be a surplus controller. Fig. 337 shows how easily the control can be arranged to act as a reducing control. It merely entails putting the vent hole on the other side of the fork.

542. DETECTION—ARCA. Fig. 338 shows part of an Arca controller using water as the operating medium. The pressure impulse enters the bellows chamber at the top left. The rocker fulcrum is fixed, but the point which receives the impulse pressure from the bellows is adjustable. The impulse pressure is balanced by an adjustable spring. The vent jet is very small and is closed or throttled by a renewable pad on the end of the rocker. The jet interferes slightly with the impulse and, in the arrangement shown in Fig. 338, acts in the same direction as the impulse. This interference is so small that it can be ignored, because the detector operates the regulator through a relay. The detector as shown is arranged to receive a reducing impulse.

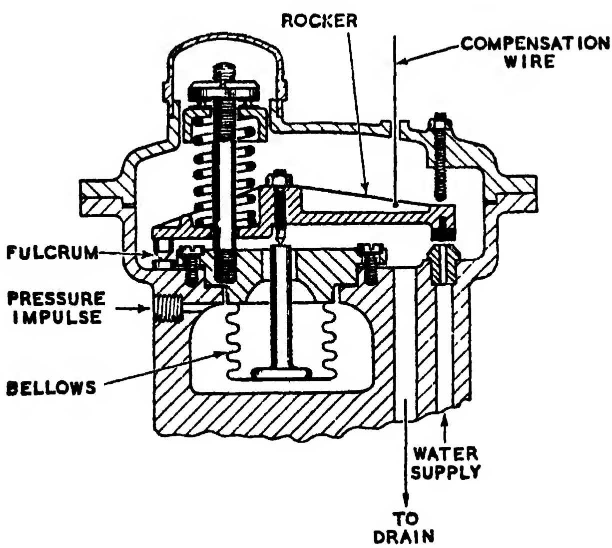

FIG. 338. ARCA IMPULSE DETECTOR

543. DETECTION—KENT. The Kent controller is particularly suited for operating any control which is to effect regulation in accordance with flow variations of any kind. The controller is simply a part of a standard Kent flow meter. The control shown diagramatically in Fig. 339 uses compressed air as the operating medium.

A meter of the U-tube type is connected to the pipe whose flow is to be used as the impulse. One arm of the U-tube is enlarged and contains a large heavy float. Minute differences of mercury level are detected by the float and multiplied by a lever system to work a chopper which passes through a gap in the air pipe. When the chopper is raised the air blows across the gap into the relay impulsing pipe. As the chopper partly or wholly chops off the air stream the air pressure in the relay impulsing pipe is reduced and the regulator is suitably activated. By shaping the chopper and by arranging that either its top edge or bottom edge does the chopping the desired characteristics can be secured.

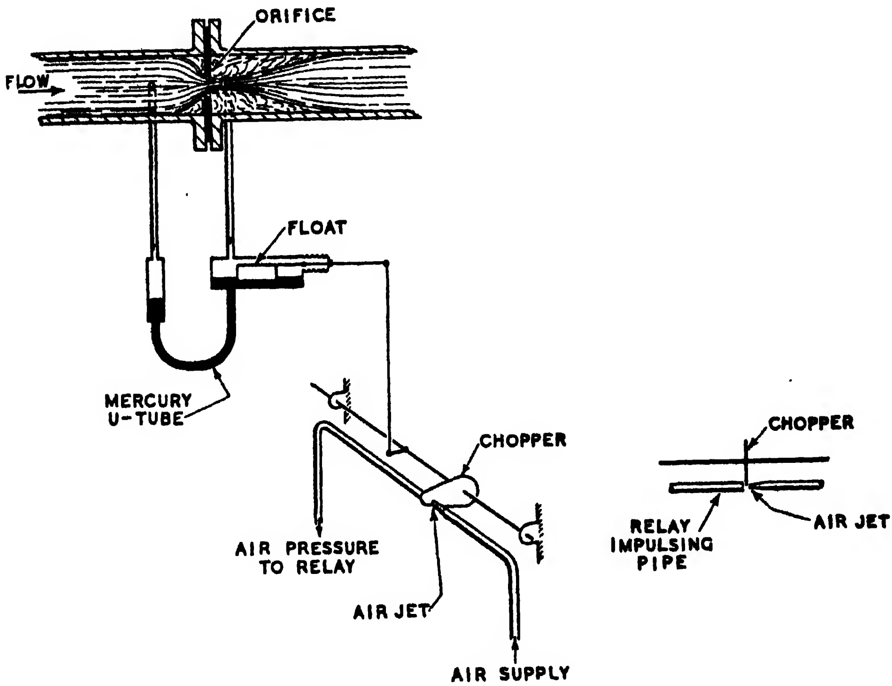

FIG. 339. KENT IMPULSE DETECTOR

544. DETECTION—OTHERS. There are many other different kinds of detector. The Electroflo uses diaphragms for pressure impulses and mercury U-tubes for flow, but the detection of the mercury level is done by a series of rods of graduated length which dip into the mercury and make electrical contacts. The Askania uses an oil jet passing across a gap as in the air gap of the Kent, but, instead of a chopper, a swivelling vane is used to deflect the oil stream. There is no need to describe all makes ; the principles are in general the same.

545. AMPLIFICATION. All high-grade controllers use a relay or servo-motor to amplify the message received by the detector. This enables the detector to be much smaller, more sensitive and cheaper and enables the

impulse to be much smaller or the regulator to be much more powerful. The amplification of electrical impulses need not be discussed; there are many methods, relay, valve, transformer. The amplification of impulses which are pressure impulses or which convert the impulse into pressure is almost always done by means of a pilot valve exactly like the valve in the reversing gear shown in Fig. 326, Section 529. Some controls use two relays in series. Some relays, such as the Hagan, are really detectors or modifiers, not true relays. The Hagan transforming relay is a combined detector and modifier and can do all kinds of complex things. The secondary impulse from the transforming relay then goes to a true relay where a tertiary impulse is sent out to the regulator. The operation of a six-chamber Hagan relay will be described later in Section 554.

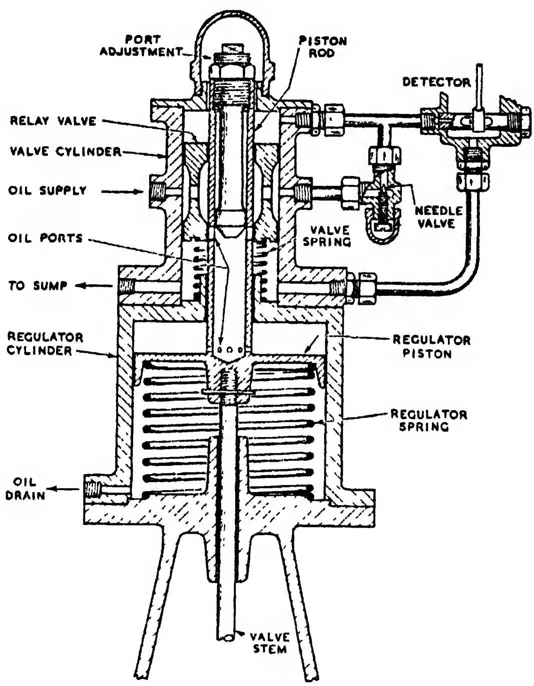

FIG. 340. RUTHS HYDRAULIC AMPLIFIER

A typical piston valve relay, the Ruths, is shown in Fig. 340, and is a simple follow up device like Fig. 326 only, by combining the relay or pilot valve with the main regulator it is possible to dispense with all external links and levers. In the actual Ruths regulator the impulse detector is carried in the same casting as the relay and regulator, but they are shown separately in Fig. 340 for clarity.

The oil supply enters the relay valve cylinder on the left and passes freely round the valve. A connection on the right hand side leads to the detector through a throttling needle valve. When the rocker fork throttles the vent, the oil pressure rises in the top of the relay valve cylinder which is connected direct to the detector. The valve will be pushed down against the valve spring.

The valve is an inside admission piston valve. The oil has free entry to the inside of the valve through the large holes shown. The valve is free to slide up and down on the hollow piston rod. The lower part only of the valve is the operating part. Its inner face covers holes in the piston rod. The amount of possible oil flow through these holes is adjustable by the screwed rod labelled "port adjustment". When the valve is pushed down oil is able to enter the centre of the piston rod. The oil passes down inside the rod and out of the holes into the top of the regulator cylinder. This oil pushes the regulator piston down against the regulator spring and opens or closes the valve that is to be regulated.

As the piston moves down it carries the piston rod with it and this movement closes the relay valve ports. The piston will move no further until it receives a further oil impulse. The action is exactly the same as the valve gear in the reversing gear shown in Fig. 326. If the detector receives an impulse such as to make the rocker fork open the vent, the valve spring can push the valve up and force the oil in the top of the valve cylinder out of the vent. The lower end of the valve now uncovers the ports in the piston rod and allows the oil in the regulator cylinder to escape through the drain back to the sump. The regulator spring pushes the regulator piston up and this closes the ports in the piston rod by sliding them under the valve. The valve and regulator piston are perfectly compensated, but this is purely a local compensation on the secondary impulse. It has nothing to do with any compensation between the regulator and the primary impulse. This relay works very smoothly and positively.

Most other makes of pilot valve or relay work on the same general principle. In some makes the pilot valve is built into the detector and the regulator cylinder may be some distance away. This calls for other means of compensation between pilot valve and regulator. This secondary compensation must be provided, because if it were not, any opening of the pilot valve would make the regulator piston travel the full stroke until the primary impulse and the detector sent it back again and the controller would hunt over its full range.

* * *

To describe all the various makes of automatic control would take too long and is quite unnecessary. Most makers of automatic controls can provide controllers to carry out similar duties to any other controller. All the reputable makes are reliable and beautifully made and there is little to choose between them.

Before continuing with the description of some automatic control applications there are two kinds that must be briefly mentioned.

546. THE SELF-BALANCING WHEATSTONE BRIDGE. These beautiful instruments—the Kent Multelec and the Leeds and Northrup Micromax—can detect, record and amplify or regulate on minute electrical impulses. This may be exceedingly useful. They are very effective, very reliable, but are expensive intricate machines.

The impulse is made to take the form of a millicurrent in one side of a Wheatstone bridge or a potentiometer bridge. The detector is a delicate

galvanometer. Now an accurate galvanometer is extremely fragile and has barely sufficient power to move its own needle let alone operate some detecting gear. The galvanometer needle in the instrument is allowed to take up its own position freely. A guillotine then drops and locks the needle. Two feelers then approach the needle scissors-fashion from either side. If the needle is central nothing happens. If the needle is deflected, due to an impulse, one feeler stops prematurely. The arrested feeler is then clutched to a rheostat. The rheostat is connected into the complementary arm of the bridge to that occupied by the impulse. The needle is then freed and the clutched feeler is centralised. The feeler that is clutched to the rheostat moves the rheostat during centralisation, by an amount and in a direction such as to bring the galvanometer back to zero because the greater or less resistance will have put the bridge in balance.

During centralisation of the feelers the centralising gear can traverse the pen on the recording chart and can also operate any secondary impulse to work a regulator.

This type of controller is particularly suitable for detecting and controlling by the minute currents from electrical thermometers or pyrometers. They are in use in the author's factory for operating on impulses of the boiling point elevation of sugar solutions.

547. CHARGE AND DISCHARGE OF CONDENSERS. The electrical controllers which detect the impulse by using it to charge or discharge condensers have some special qualities that make them particularly suitable for certain applications. They can sort out a number of impulses and can add or subtract these impulses. They are particularly suited to giving variable time lags. An example will be described using the application that is most familiar, namely the operation of traffic lights at a cross road. It is impossible to spare the space to describe the controller in detail and the diagram used is a skeleton diagram only, the real circuits are too complicated for quick understanding, except by an electrician.

A condenser C in Fig. 341 is charged by the supply current through an adjustable resistance B. The rate of charge is so adjusted, by altering the resistance B, that the condenser will not be fully charged until a sufficient time has elapsed as will permit a vehicle to start and get well over the cross road. The condenser can be discharged through either of two circuits. The left hand circuit is connected through the adjustable resistance D to the road contacts G carrying the green right of way. The resistance D is adjusted to suit the speed of the traffic and the width of the cross road. Each time a vehicle crosses the contacts G the condenser C is partly discharged. If the vehicle is going very slowly the condenser is almost completely discharged due to the length of time that the slow vehicle will make the contacts G. If the vehicle is going very fast it will only need a short right of way time and this is automatically secured because the short time that the fast vehicle made the contacts G will only discharge the condenser a small amount.

As soon as there is a lull in the traffic the condenser C will get fully charged. If now a vehicle makes contacts R on the red road, the condenser C will discharge through the relay E. This relay impulses the regulator which initiates

the sequence amber, green, etc. The neon lamp F is inserted in the right hand circuit so that this circuit cannot pass a current until the condenser is fully charged so that there will be a sufficient current flow to operate relay E smartly. During ionisation in the neon lamp no current passes, but as soon as the lamp glows the current flows fully.

Clearly the controller as so far described is imperfect. A vehicle would only succeed in impulsing the controller if it happened to make contacts R when the condenser was fully charged. This difficulty is surmounted by the contactor H, whose local circuit is omitted for simplicity. When a vehicle makes contacts R it locks up contacts H and H′ and so stores up in the controller a message that

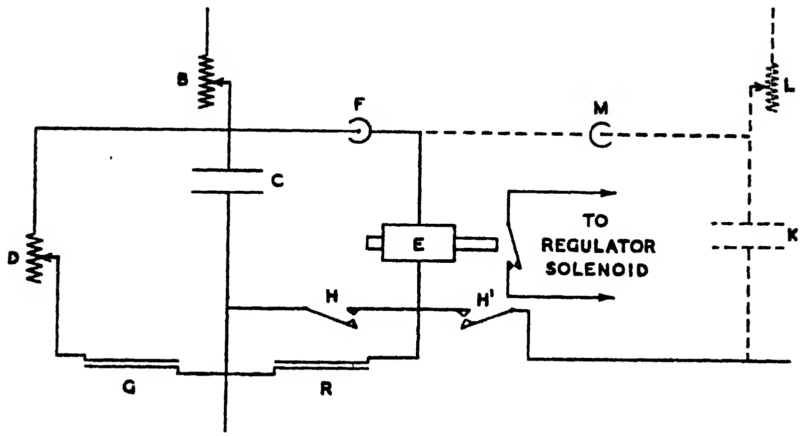

FIG. 341. TRAFFIC OPERATED LIGHT CONTROLLER

a vehicle is waiting at the red light. As soon as the condenser C is sufficiently charged to discharge through the neon lamp, this discharge will take place through contacts H. In other words a vehicle approaching the red light says to the controller " Let me through ", and the controller does so as soon as there is a sufficient lull in the other traffic.

But there is still an imperfection. If there is a continuous stream of traffic across the green right of way the condenser C will never get charged, and the vehicle waiting at the red light will have to wait indefinitely. This difficulty is overcome by the dotted circuit. The condenser K is charged very slowly through the very high adjustable resistance L. As soon as this condenser is fully charged it will discharge through the neon lamp M and the relay E if the contacts H′ are made. This means that after a certain adjustable period of right of way on the green road the regulator will change the lights if a vehicle is waiting or arrives at the red light, regardless of the density of the traffic on the right of way.

As soon as the controller operates relay E it starts a sequence of contact changes to put the green light to amber and the red light to amber and red, etc. In the new position when the side road has the green right of way the time lags

must all be different. So that the last operation of the regulator is to put a new set of resistances into operation and to change the road contacts G and R to opposite circuits.

All kinds of refinements can be and are introduced. When the condenser C discharges and the regulator puts the green light to amber, the amber period is controlled by the charging of another condenser. If this condenser is charged rapidly through a low resistance the amber period will be short. By allowing contacts G to discharge the amber condenser to a small extent the amber period can be lengthened. This is done in some cases so that a driver racing up to beat the lights puts in an " amber extension " and is thus protected against himself.

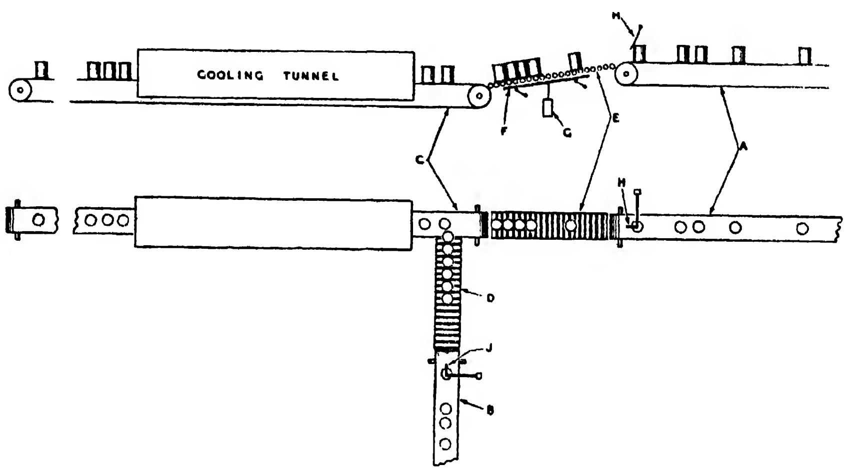

FIG. 342. CONDENSER CHARGE CONTROLLER APPLIED TO COOLING OF FOOD CANS

This condenser charging and discharging system is obviously an admirable way of adding and subtracting impulses and of providing all kinds of adjustable time delays. It is not confined to traffic control. Fig. 342 shows an application to a food packing problem. Cans are filled with a hot material which must be cooled as soon as possible before being stored in warehouse. The cans are filled at two points and with material at different temperatures and are placed on the steel bands A and B. The output from both conveyors goes to conveyor C which passes the hot cans through a refrigerated cooling tunnel. Cans from conveyors A and B are transferred to conveyor C by the gravity rollers D and E. These gravity rollers are provided with brakes F operated by thrusters G. The detection fingers H and J impulse the traffic controller which applies the appropriate brake to the gravity rollers of the output of cans, thus economising in refrigerating power. In order again to lighten the load on the refrigerator the fingers H and J give an overriding prophetic impulse to the thermostatic controller operating the refrigerator. If the temperature of the cans is substantially different on either band the hot cans can very easily give a larger impulse to the refrigerator regulator by operating through a lower resistance. This enables the refrigerator to anticipate the arrival of many hot or few tepid cans.

The condenser controllers are obviously particularly suitable for counting or controlling a series of single happenings, they are not so suited to controlling the variations in a bulk material.

These digressions were impulsed by the consideration of multiple impulses. Before we continue our examination of some actual controllers in detail there are two matters which require a little attention. The first is the ensuring that the impulse will be correctly transmitted to the detector; the second is the modifications that can be applied to a compensating gear.

548. CORRECT IMPULSE. All controllers depend fundamentally on their impulse. Unless the impulse delivers the right message regarding the state of affairs to be controlled, the controller, be it never so tricky, cannot possibly do its task correctly. When the impulse consists of steam pressure there is always the risk of condensate collecting in the impulse pipe and either adding its hydrostatic head to the impulse or locking the impulse pipe and strangling the impulse. The first danger occurs if the controller is below the pipes from which the impulse is being taken; the second danger occurs if the detector is above the impulsing pipe.

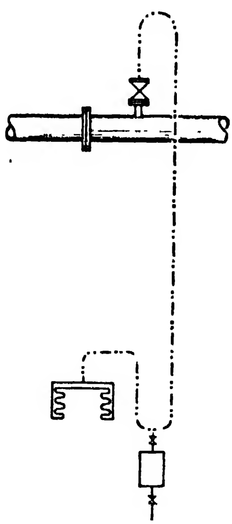

FIG. 343. ENSURING CORRECT PRESSURE IMPULSE WITH CONTROLLER BELOW STEAM PIPE

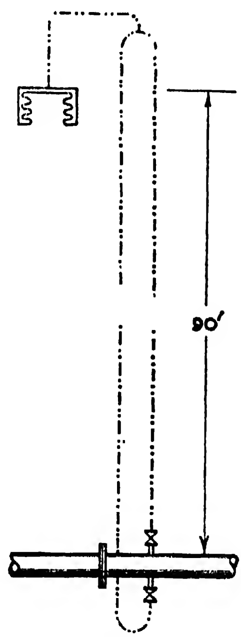

FIG. 344. ENSURING CORRECT PRESSURE IMPULSE WITH CONTROLLER ABOVE STEAM PIPE

Fig. 343 shows a detector fitted below the impulsing pipe. The impulse should be taken off the top of the pipe to reduce the possibility of the impulse

pipe filling with condensate. The impulse pipe is brought down to a level just below the detector and carries at the lowest point a condensate collecting pot—which must not be fitted with an intermittent trap lest the blast action interfere with the impulse, but can carry a very small continuous trap. If there is only one impulse, or if more than one impulse comes from the same level, it may be permissible to keep the impulse pipes full of water. The precautions that have to be observed when the impulse pipes are kept full have been dealt with in Sections 224 and 227 in Chapter 6. It may sometimes be possible to keep the impulse pipes full of air.

Fig. 344 shows the arrangement adopted in the author's factory for keeping steam pressure impulse pipes clear of condensate where the detector is some 90 feet above the pipe whose pressure is giving the impulse. Two impulse pipes are taken off the top and bottom respectively of the main and the actual impulse is taken off the top of the loop formed by the two impulse pipes. This arrangement induces a minute steam circulation in the impulse loop and prevents any accumulation of condensate except in that part of the impulse pipe below the main. This arrangement has proved satisfactory on superheated steam.

549. MODIFIED COMPENSATION—ANGLING BAR. The direct compensations shown in Figs. 129, 130 and 334 are simple and straightforward and give linear or directly proportional compensation. Now in some cases it may be desirable to give a lot of compensation at one end of the regulator's movement and very little the other. Or, it may be desirable that the compensation should be more intense at one particular part of the regulator position. Any such effects can be very easily obtained.

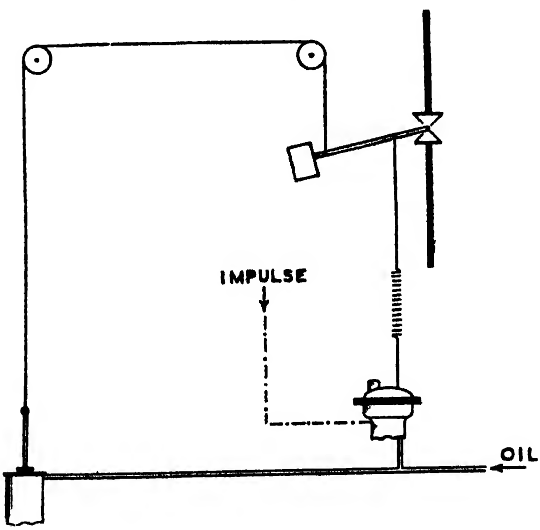

FIG. 345. STRAIGHT LINE COMPENSATION WITH WIRE AND SPRING

Fig. 345 shows a detector controlling a regulator which operates a valve. Direct straightline compensation between valve movement and detector is furnished by the wire and spring shown. Modified compensation is much more

easily secured by adding an angling bar to the wire and spring. Fig. 346 shows the angling bar whose action is self-evident without explanation. In Fig. 346a the angling bar is so curved as to give an exact straight line compensation identical with that produced by the wire in Fig. 345. The straight angling bar shown in Fig. 346b gives an ever diminishing compensation as the valve opens. In Fig. 346c compensation is cut out altogether during all the middle travel of the valve and very quick compensation is applied near the full open and full shut positions. In Fig. 346d the opposite arrangement is shown. There is no compensation near the full shut and full open positions but considerable compensation all over the main travel of the valve.

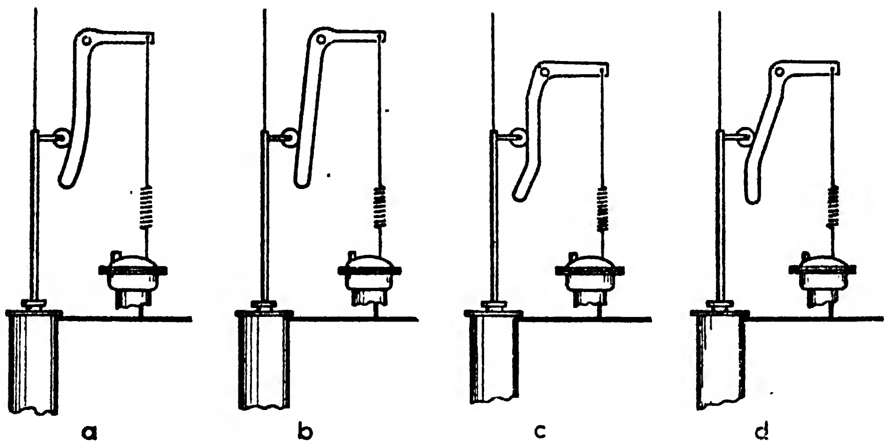

FIG. 346. MODIFIED COMPENSATION WITH ANGLING BARS OF VARIOUS SHAPES

By modifying the compensation it is often possible to make the controller give all kinds of tricky effects, either as a compensator or by using the regulating cylinder and angling bar as a transforming relay to produce a curved characteristic from a straight line impulse. Although it may be desirable to give a controller straight line compensation, it is almost always impossible to do this in practice by means of the plain wire connection, because the valve characteristics are seldom straight line functions. For this reason it is generally better to use an angling bar as a compensator rather than a plain wire connection. In some automatic controllers the valve ports are so cut that the valve gives a substantially straight line flow/opening characteristic. In such cases the plain wire will be quite satisfactory, as it will be if compensation is not of very great importance.

We can now return to the consideration of actual controllers. The control described in the following Section is complicated and deals with a somewhat tricky steam pressure problem. As it requires some concentration to follow, those readers who are not interested in the control of several interdependent steam pressures are advised to skip it, but they are asked to finish the remaining sections of this chapter which deal with the very important problem of the control of tanks with which this chapter opened.

550. PASS-OUT, SURPLUS AND REDUCING VALVE CONTROLLER. This is a more complicated system than any yet considered. It has been chosen to show how easily a number of impulses can be led to a controller, sorted out by simple combinations of detectors and translated into the action of two regulators. The controller contains seven detectors, five of which are compensated. The actual controller is very small, compact and simple although it appears complicated. It was designed and made in the author's factory using standard Arca parts. The operating medium is oil.

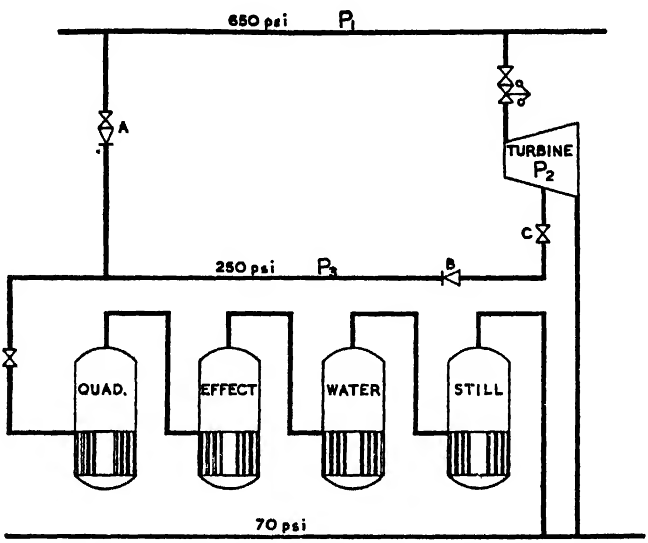

FIG. 347. PASS-OUT, SURPLUS AND REDUCING CONTROL—THE PROBLEM

Fig. 347 is a skeleton diagram showing the problem. Steam at 650 psi is supplied to the back pressure turbine which exhausts to the process at 70 psi. Distilled water for boiler feed is obtained from the quadruple effect still working between 250 psi and the process main at 70 psi. The steam for the still is normally drawn from the pass-out connection on the turbine. When the turbine cannot supply sufficient pass-out steam for the still, make up 250 psi steam can be obtained from the 650 psi main through the reducing valve A. The object of the controller is to provide the maximum of power and the maximum of distilled water with the least possible use of the reducing valve A.

When the turbine is on nearly full load no pass-out steam can be taken. When the boiler pressure is low no steam must be taken through the reducing valve A lest the turbine cannot produce enough power with the lower pressure. No steam must be taken through the reducing valve A if it can be taken from the pass-out point. To ensure this the reducing valve A must not open until the pressure P_3 on the 250 psi main is more than 10 psi lower than the pressure P_2 at the turbine pass-out point. The pass-out valve must be shut during the process of putting the turbine on or off load, that is to say when the pass-out pressure P_2 is below 150 psi.

We can tabulate the requirements thus :—

	Regulator	P_1	P_2	P_3
B	Shut	.. ..	150 or less ..	260 or more
B	Full open	.. ..	180 or more	240 or less
A	Shut..	615 or less ..	Less than 10 psi above P_3.	240 or more
A	Full open	More than 615	More than 10 psi above P_3.	220 or less
B	Shut..	When the turbine is governing on the last valve.		
A	Must operate as a simple surplus and reducing valve when the turbine pass-out stop valve C is shut.			

The last requirement is met by means of a simple on-off impulse from an electric switch on valve C. The last but one requirement is so rare, and the turbine driver will anyhow be so much on his toes in these circumstances, that this operation can be done by manual operation of stop valve C by the turbine driver.

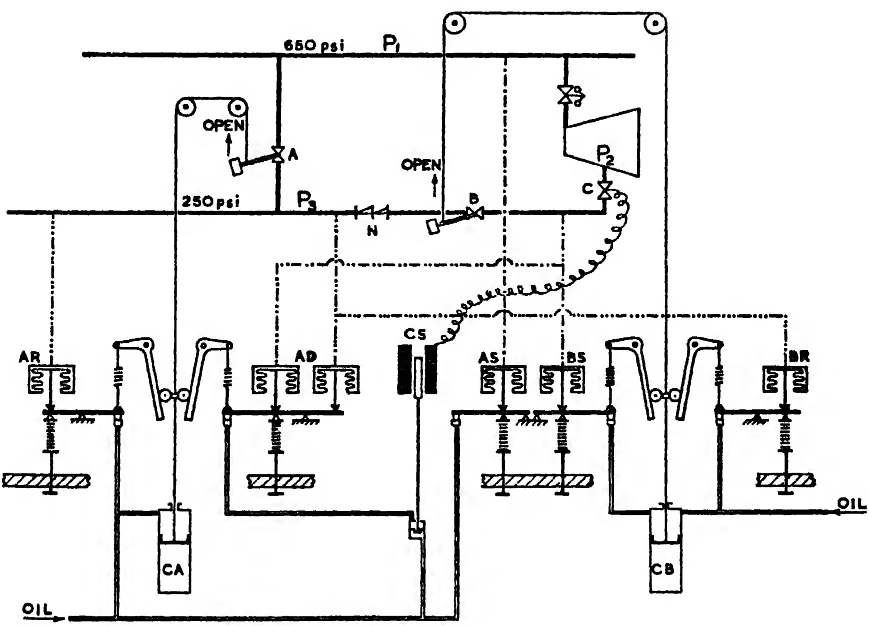

FIG. 348. PASS-OUT, SURPLUS AND REDUCING CONTROL—THE SOLUTION

Fig. 348 shows the controller in diagrammatic form. As shown the detectors apply the oil pressure directly to the regulating cylinders. This is only shown thus for simplicity. In practice the secondary oil impulses from the detectors operate the pilot valves of relays on the regulating cylinders. This simplification makes no difference to the general action of the device.

Cylinder CA regulates the 650/250 reducing valve A. This is an emergency valve which is a last resort. It is controlled by the reducing control AR, the differential control AD and the surplus control AS. Whenever the pressure P_3 is below 240 psi the detector AR throttles its vent in an endeavour to open valve A, and at 220 psi will completely close the vent and call for full opening of valve A. It cannot do anything however unless the other controls AD and AS are also demanding an opening of valve A. The surplus detector AS opens the vent fully if the 650 psi pressure falls below 615 and gradually throttles the vent above 615. So that the valve A cannot open unless P_3 is below 240 and P_1 is above 615 psi. The differential detector AD is in balance when the left-hand chamber, impulsed by P_2 is 10 psi higher than the right-hand chamber, impulsed by P_3. Consequently when P_3 drops to more than 10 psi below P_2 the valve A will open, provided the surplus and reducing impulses will permit. When P_3 rises to 10 psi or less below P_2 the detector will open its vent and prevent the opening of valve A regardless of any surplus or reducing impulses, received by AR or AS.

Now when the pass-out stop valve C is closed, the valve A must operate as a straight surplus and reducing valve, so that the differential control AD must be put out of action. This is done by the solenoid CS which closes the little valve in the oil supply to the AD vent.

The control has thus fulfilled the conditions of giving P_3 the minimum of P steam when there is P_2 steam available.

Valve B, the pass-out valve, is controlled by BS which is the surplus control impulsed by the turbine pass-out pressure, and BR which is the reducing control impulsed by the 250 psi main. BS is set to call for a full opening of valve B as soon as the turbine pass-out pressure is above 180 psi. This control closes valve B whenever the turbine pass-out pressure drops below 150 psi. BR is the reducing control and tries to open valve B fully whenever the still pressure P_3 is at 240 psi or less. When the P_3 pressure is at 260 psi or more the control BR closes valve B.

Controls AR, AD, BS and BR are compensated by means of angling bars. Any of them may at any time be doing the controlling. Control AS is not compensated because it is not a regulating controller but a permissive on-off controller which merely permits the controls AR or AD to regulate valve A.

At N in Fig. 348 are two non-return valves in series. These are of different types and different makes. Their object is to protect the turbine in the event of the generator tripping out with the pass-out in action, lest the turbine were supplied with steam from the 250 psi line on no load and might run away, because the governor has no control over the pass-out.

It requires some concentration to grasp the working of this controller, but in actual practice it is exceedingly simple and does its job. It fulfills all the conditions laid down in the stiff specification. The valves and Arca components cost £612 in 1943. The installation labour £210, and draughtsman's time was about 150 hours.

551. TANK CONTROL. Having now seen the kind of things automatic controls can do and having investigated the details and operations of various kinds of controls we can return to the original problem with which we started

this chapter. How can a tank be controlled so that it fulfils its functions and acts as a tank to smooth out irregular processing ?

We will assume that the control is to operate on the inflow to a tank and that it is to meet as far as possible the following set of conditions :

The tank is to smooth out fluctuations in outflow with the minimum of interference to inflow.

The full capacity of the tank is to be used.

The inflow must be checked when the level is high and rising.

The inflow must be fully opened when the level is low and falling.

The control must ignore a high but falling level.

The control must ignore a low and rising level.

The control must allow as much freedom as possible about the average flow.

The control must adjust itself to changing average flows.

This very stiff specification is met, for all practical purposes, by two controllers which will shortly be described. It might be thought that it can never pay to instal such a refined instrument for such a mundane task as the control of a process tank. The answer is that the price paid for fluctuations must be worked out, as must the price paid for hold-ups. These must then be compared with the cost of the controller.

If the plant is of a certain size and the throughput is fluctuating, obviously a greater output could be obtained were the throughput steady. The plant cannot do more than maximum throughput. When the throughput is fluctuating the maximum only occurs at times of peak. With continuous running the throughput can be set at just below, or even at, maximum all the time. Fluctuations in throughput can easily degenerate into intermittent running, which is even worse because it entails dead stops. The cost of dead stops must be calculated and their duration must be measured. In the author's factory in 1957 a dead stop cost about two shillings a second.

Suppose maximum throughput is not required. What then is the value of smoothness of throughput ? As an example, not necessarily typical but definitely existent, let us take the case of the author's factory. When output is below maximum the density of the sugar liquor is so increased—to avoid subsequent evaporation—that the liquor will just process and no more. If the throughput varies there must be many periods of over average running to compensate for the valleys of under average running. This means that the liquor density must be set so that the peak rates can be processed. It is probably no exaggeration to say that perfectly smooth working would permit of processing at 1 per cent. higher density than when the throughput fluctuates.

30 per cent. water for 12,000 tons sugar weekly is	5,145 tons of water.
31 per cent. water for 12,000 tons sugar weekly is	5,385 tons of water.
Extra evaporation..	240 tons of water.
or	30 tons of coal a week
at 100*s*.	£7,500 a year.

It may therefore pay handsomely to spend time and money on tank control, if the throughput is large enough.

If our tank controller is to fulfil the conditions there must be no direct compensation between tank level and valve opening. At times the tank level may be high and the valve may be either full open or tight shut. One of the reasons for the failure of the simple float valve is that there is perfect compensation between level and valve opening. So that either suppression must be used or compensation must be applied separately between the impulse and the detector, and between the controller and the regulator. The controller first described uses suppression, while the second controller uses double independent compensation.

552. MODIFIED IMPULSE. We want the controller to ignore broadly any variations in level throughout all the middle ranges of tank level. We want it to ignore a high level if the level is falling, and to ignore a low level if the level is rising. We want it to adjust itself approximately to the temporary average rate of flow. The controller must be especially on its toes when the level is very high or very low. Clearly there must be two kinds of impulse ; a level impulse and a rate of change and direction impulse. The level impulse must be violent near the full and empty levels but must be negligible throughout the main range. It is possible to make the device that produces the impulse produce a modified impulse that fulfils this need.

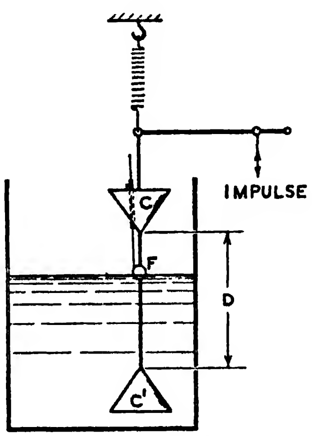

FIG. 349. MODIFIED IMPULSE WITH SHAPED DISPLACER. ADJUSTABLE BALL GIVES SMALL RESTORING OR RESETTING IMPULSE

As we want some sort of impulse over the whole range of tank level it will probably be best to use a displacer. If we use a displacer we can shape it like that shown in Fig. 349. This displacer would consist of a light, say ¼ in. rod with large conical displacers fitted to the top and bottom which will give increasingly powerful impulses as the liquid level approaches the top or bottom of the tank. Over the bulk of the tank range D the displacement of the ¼ in. rod is so small as hardly to affect the impulse at all.

Sliding on the rod between the conical displacers is the ball F which is attached to a stout wire by which the height of the ball can be adjusted and locked to the top cone C. If we so adjust the controller that it is in equilibrium when C′ + ½F are submerged, the control will always try to bring the tank level back to the centre of F wherever F may have been fixed. If F is made very small relative to C and C′ the tendency of the controller to restore the level to F will be slow. The effect of ball F is to bias the controller to restore the tank level to the level at which F has been set and thus to adjust itself to changing average rates of flow. If F were omitted there would be nothing to prevent the controller operating either on C or C′ with the tank either full or empty if the average flow rate changed. By giving the controller a bias towards restoring a half full tank the controller will adjust itself to permit the use of the whole tank for fluctuations about a changing average. This modification of the impulse at the very start has greatly simplified the task of the controller in the simplest possible way.

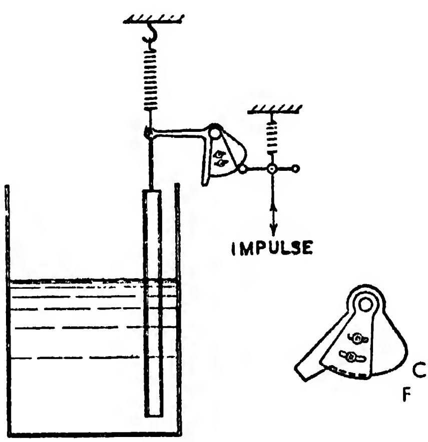

FIG. 350. PARALLEL DISPLACER AND CAM CAN GIVE IDENTICAL IMPULSE TO THAT FROM THE DISPLACER IN FIG. 349

The same result can be secured from a parallel displacer by using the device shown in Fig. 350. The parallel displacer gives a straightline movement to the displacer which can send out a modified impulse through a suitably shaped cam. The displacer moves by an amount directly proportional to the change in level and rocks the cam which moves the impulse lever. The steep cam faces C and C′ transform the straight impulse given by the displacer into magnified and increasing impulses identical with those produced by the conical displacers in Fig. 349. The position of the step F in the cam can be adjusted and this step corresponds exactly with the ball F on the displacer in Fig. 349.

We have now got two ways of giving an impulse to the controller which is to all intents and purposes constant over the middle range of tank levels, which increases just as fast as we like to arrange it near the top and bottom of the tank and which gives a small continuous message to the controller demanding a general restoration towards an adjustable mid point.

553. ELECTRICAL DETECTION AND CONTROL. Two of the essential requirements of the tank controller are that it should ignore a low level if the level is rising and ignore a high level if the level is falling, and that these ignorings should be varied at very high or very low levels. That is to say, if the level is very high the message must only be ignored if the level is falling very rapidly, and if the level is very low action must result unless the level is rising very rapidly. Now this is exactly what the Kent master boiler controller does.

Fig. 351 shows diagramatically the essentials of a Kent master controller applied to tank level control. The impulse is derived from the shaped displacer A so that a suitably modified impulse is given. Changes of level operate the contact rockers B and C. Rocker B moves the contact D in exact reduced proportion to the movement of the displacer. The tension of spring E and the volume of the displacer are so adjusted that the contact D is central, that is on line *b f*, when the liquid level is just half way up the displacer ball F.

Every 15 seconds the scissors contacts G and H, approach contact D under the action of cam J and spring K. These contacts detect whether the tank level is high or low, making a very accurate and magnified detection near the top and bottom of the tank and a very small detection over all the middle range.

Rocker C is coupled to the impulse link L through a light friction clutch. The leverage is such that, for a given impulse, contact M moves much more than contact D. When the scissors contacts P and Q approach contact M by the action of cam N and spring O they not only detect the position of contact M but they centralise it. So that at each 15 second cycle contact M records the amount and direction of the change of impulse during the previous 10 seconds. When the tank level rises the contacts D and M fall. Contacts D and M are connected to the D.C. supply and contacts G P and H Q are connected to either side of the split field motor R.

When either contact D or contact M connects with G or H or with P or Q the motor will start in the appropriate direction. The length of time that the motor runs depends on the time that the contacts are made. The time that the contacts D G H make depends on the tank level. The time that contacts M P Q make depends on the change of tank level during the last 10 seconds. The motor R operates the valve through suitable reduction gearing.

Now suppose the level rises to any level F″ above F′ but below A′. Contact D will be displaced downwards a very small amount due to the displacement of ½ F. Contact M will also be displaced downwards by a rather larger amount. Contacts M and Q will make and motor R will slightly close the valve S. Contact M will be centralised and will take no further part in the proceedings so long as the tank level remains between F″ and A′. At the next detection movement of the cam mechanism contacts M P Q will not make, but contacts D H will make for a very short time and will continue to move the valve S by tiny amounts in an endeavour to restore the level to F′—in other words to set the controller to the temporary average rate of flow.

Now suppose the level rises to A″. Contact D will be considerably displaced downwards and contact M will be displaced downwards very much more. Contacts M Q will give a long close impulse to the valve motor. If the level

remains at A″ contact M will have been centralised and will take no further part in the temporary proceedings but contacts D H will make for a moderately long time and give longish closing impulses to the valve motor.

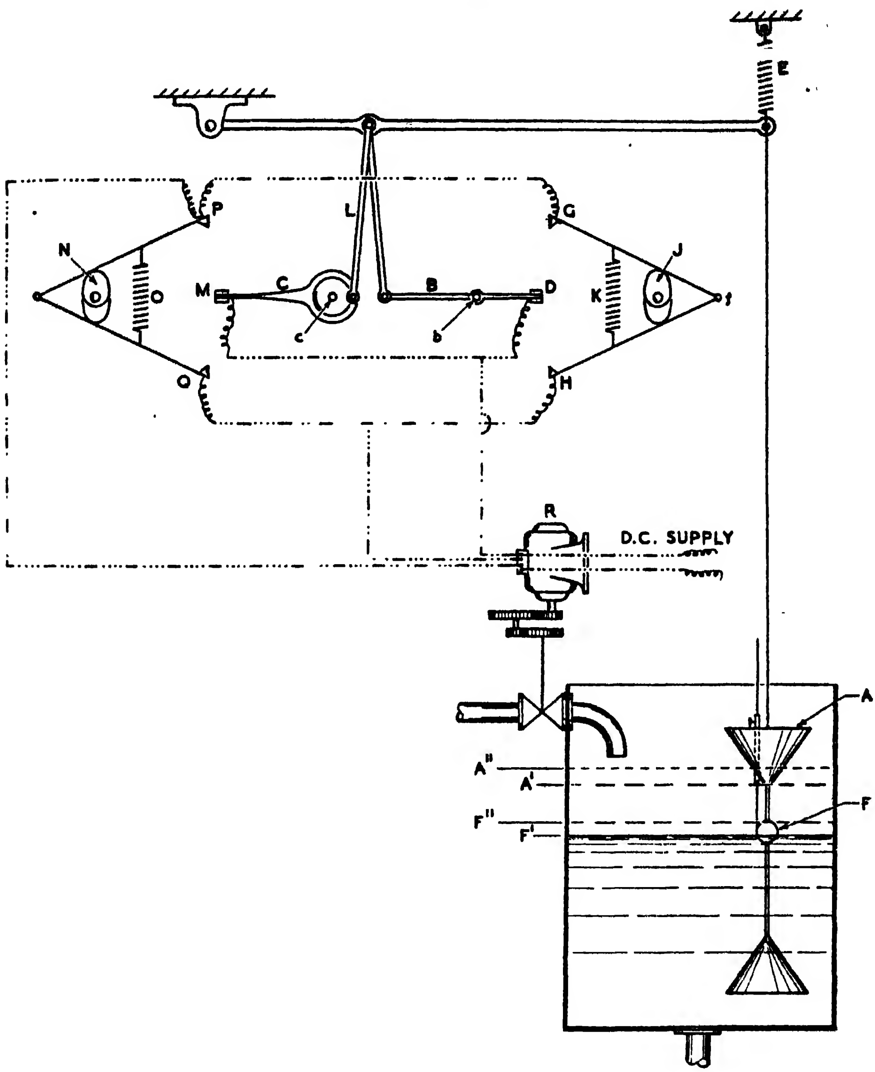

FIG. 351. KENT MASTER BOILER CONTROLLER APPLIED TO TANK LEVEL CONTROL

As soon as the valve has closed sufficiently to cause the level to fall below A″ contact D will rise slightly and will therefore try to give a smaller " close " message, but contact M will have risen quite a lot and, making contact with P will not only cancel the D H " close " message but may give an " open " message. In this way the controller can ignore or even reverse a high level message if the level is falling rapidly or do the same to a low level message if the level is rising rapidly.

This control provides a complete solution to the tank control problem. Hunting is eliminated and all the specified tasks are carried out. There is no compensation but there is complete suppression.

554. TANK CONTROL—PNEUMATIC IMPULSE CONVERSION. A similar result, possibly not quite so perfect in some ways but having advantages in others, can be obtained with a pneumatic control embodying a Hagan transforming relay which acts as detector and controller and does the same things as the Kent electrical device. This transforming relay looks very complicated in a drawing, but is not really difficult to follow if the six chambers are considered separately or in pairs.

Fig. 352 shows the whole control in exaggerated scale for clarity. The displacer 1 gives a modified mechanical impulse to the lever 2. This modified mechanical impulse is transferred to the plunger 3 of the impulse converter. The impulse converter is simply a reducing valve which converts the mechanical message from the displacer into air pressure. The spring 4 in the converter is compressed or released by the movement of the displacer, and its increased or decreased pressure acts on the diaphragm 5. The diaphragm 5 carries a valve seat 6 which also acts as a bolt to tie the diaphragm assembly together. The air valve 7 is held against the seating 6 by the light spring 8. The other, lower end of valve 7 ends in a ball which is lifted towards the seating 9 by the spring 8. When the diaphragm is in equilibrium valve 7 is seated on both the seatings 6 and 9. When the diaphragm is depressed, the seating 6 pushes the valve 7 down and opens the valve at 9. When the diaphragm rises, valve 7 seats on the seating 9 and opens at 6. Seating 6 is an exhaust valve to atmosphere through the hole 10. Seating 9 is the inlet from the compressed air supply. The valve 7 and its seatings have been shown greatly enlarged for clarity.

From this arrangement it is clear that the spring pressure causes the impulse converter always to produce such an air pressure under the diaphragm in chamber A as exactly to balance the spring pressure. In other words this reducing-valve-converter translates a mechanical displacer impulse into an air pressure impulse.

555. TRANSFORMING RELAY. The Hagan transforming relay consists of six chambers B, C, D, E, F and G. These are separated by alternate flexible and rigid diaphragms. The centres of the moving diaphragms are all tied together by means of a central bolt and distance pieces. Frictionless leakproof joints are provided between the movable assembly and the fixed diaphragms by the four bellows.

The air impulses from the converter enter chambers C, D and E. Assuming conditions are stable the pressures on either side of diaphragm 11 are the same, so these chambers are exerting no force on the central assembly. For the moment we will ignore chambers D and E.

The impulse pressure in chamber C is acting on the diaphragm 12 and driving it downwards against spring 13. In addition to the downward force exerted by the pressure in chamber C the spring 14 is also exerting a downward force. Spring 14 is adjustable by means of the handwheel at the top of the relay.

The displacer 1 has no levelling ball on its stem. Instead the middle stem is a rod of small but definite displacement. The air pressure from the impulse converter will have a certain definite value for every tank level. The transforming relay will be in equilibrium when spring 14 + impulse pressure in C = Spring 13. Adjusting the load on spring 14 corresponds exactly to adjusting the position of ball F in Fig. 351.

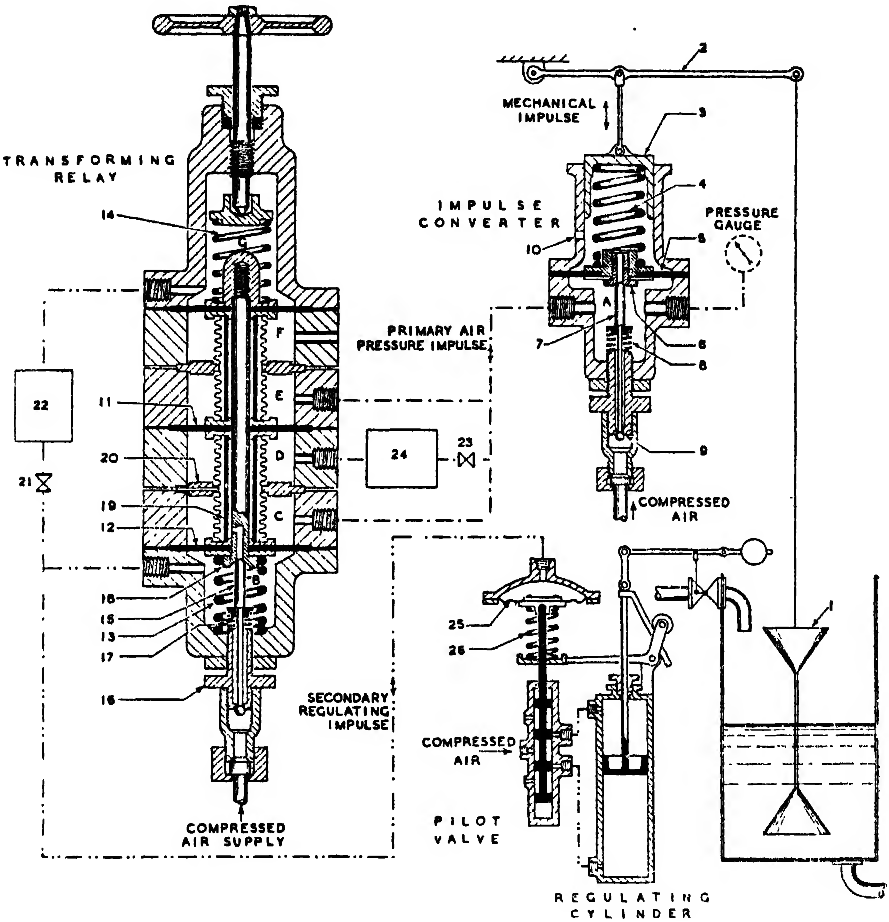

FIG. 352. HAGAN PNEUMATIC TANK LEVEL CONTROL

In chamber B is a valve 15 exactly the same as valve 7 in the impulse converter. The lower part of valve 15 ends in a ball held against the adjustable seating 16 by means of the light spring 17. Whenever the diaphragm assembly is depressed the ball end of valve 15 clears the seating 16 and compressed air is admitted to chamber B. Similarly if the diaphragm assembly rises, the ball end at the bottom of valve 15 is held on the seating 16 and the top end of the valve clears the seating 18 and allows air in chamber B to escape up the central

hole in the bolt whose lower end forms the valve seat 18. A small cross hole in this bolt allows the escape of air through the hole in the distance piece 19 and thence through the hole in the fixed diaphragm 20 to atmosphere.

Now chamber B is in communication with chamber G through the needle valve 21 and the air reservoir 22. The pressure in chamber B actuates the regulator which will be described in a moment. When the equilibrium of the diaphragm assembly is upset, a pressure change is made in chamber B by means of the valve 15. This pressure change restores the equilibrium of the diaphragm assembly and neutralises the impulse. But after a delay which is adjustable by the needle valve 21, the pressures in chambers B and G are equalised. If the diaphragm assembly is still out of equilibrium another pressure rise takes place in chamber B and another secondary impulse is sent out to the regulator. Then after a short delay the pressures in B and G are equalised and so on.

By allowing delayed equalisation of the pressures in chambers B and G exactly the same effect, of a number of secondary impulses to the regulator, is obtained as in the 15-second measurement of the impulse in the Kent system.

The transforming relay is self-compensating. The impulse pressure in chamber C actuates the diaphragm which restores equilibrium and temporarily neutralises the impulse, by adjusting the pressure in chamber B.

The action so far described corresponds exactly with the high-low detection by contacts D G H in the Kent controller in Fig. 351.

The primary impulse is connected to chambers C, D and E. The connection to D goes through a needle valve 23 and an air reservoir 24. The goings on in chamber C have been considered. We must now consider what chambers D and E are for.

If the reservoir 24 is large and the valve 23 is only open a very small amount, the pressure in chamber D will change very slowly. Chamber D can be looked upon as indicating what the value of the impulse " was ", whereas chamber E records what the value of the impulse " is ". The difference in pressure between chambers D and E shows the rate of change of the impulse and the direction in which it is changing.

If the tank level is rising, the primary impulse pressure is also rising, and the pressure in chamber E will be above that in chamber D. There will therefore be an additional pressure due to this pressure difference, acting on the diaphragm assembly and this force will add itself to the impulse pressure in chamber C.

Now suppose the tank level is high but falling. Chamber C will be continually trying to send secondary regulating impulses out to the regulator, but chamber D will be at a higher pressure than chamber E. The resulting upward force will neutralise or even exceed the impulse force in chamber E.

Chambers D and E together with reservoir 24 and valve 23 form the exact counterpart of the rate of change detector contacts M P Q in the Kent controller in Fig. 351, and form the reset part of the control.

Chamber F is a passenger. It only exists because it forms the " other side " of the top diaphragm. It serves no purpose (doubtless the ingenious could find some tricky use for it) and it communicates with the atmosphere.

Now the transforming relay is an instrument, not a regulator. It sends out secondary impulses to the diaphragm 25 which operates the regulator pilot valve. As the diaphragm 25 moves it causes the pilot valve to admit compressed air to the top or bottom of the regulating cylinder. The regulator piston rod carries an angling bar that provides compensation by tightening the spring that opposes the pressure on the diaphragm 25. As the diaphragm 25 calls for greater regulator movement the angling bar adjusts the pressure of spring 26 so as to neutralise the movement of the diaphragm 25 and so restore the pilot valve to the central position. Were this compensation not provided there would be nothing to restore the pilot valve to the mid position and the regulator would always travel the full stroke. As the pilot valve—an inside admission piston valve—opens one port to compressed air it opens the other port to exhaust to atmosphere.

The needle valves 21 and 23 and the expansion chambers 22 and 24 are replaced on Foxboro controllers by lengths of very fine bore tubing which offer so much resistance to flow as to delay the transmission of the pressure change by any desired time lag.

556. COMPARISON OF TANK CONTROLLERS. It is difficult to say which of the two tank controls, Kent or Hagan, is the best. The Hagan has been described as if its action were intermittent. Of course this is not so. The pressure in chambers B and G do not differ, equalise, differ, equalise. They veer slowly away from one another and smoothly approach again. For this reason it might be said that the Hagan was the better control. Also in its favour are no contacts, no commutator. But it has disadvantages. It is much more difficult to understand. It is difficult to see what it is doing except for small changes on two pressure gauges, whereas the Kent impulses can be made clear by pilot lights.

It will have been noticed that the impulse converter and the transforming relay are internally self-compensating. Each is compensated by itself, but there is no compensation between them. The pilot valve and regulator have another separate compensation system. There is no compensation whatever between impulse and regulator, which was one of the conditions laid down in Section 551.

All the tank controls have been described with the control regulating the inlet. There might be circumstances where the outlet should be controlled, or where there should be an added control on the inlet to accommodate varying inflow.

557. COST OF AUTOMATIC CONTROLS. Automatic controls are very expensive. This is because they have to be beautifully made and because a great deal of brains and experiment have been used in their development, but principally because so few automatic controls are used. If industries would start buying automatic controls to the extent that their usefulness justifies the price would come tumbling down and we all could install more.

* * *

There are many excellent makes of automatic control that have not been described in this Chapter—Electroflo, Foxboro, etc. This chapter is already

overlong and the controllers were described simply because most of them are at work in the author's factory, not because their merits are necessarily any greater than those of other makes.

* * *

There are many other types of automatic control, some simpler—like plain reducing valves or steam traps, some more complicated—like automatic pilots. The controllers that have been described cover a fair range and all of them save steam or power, which is steam. It is hoped that the choice of controls described has been happy and that, at any rate, the humble float valve will not in future be treated in the off-hand manner to which it has become accustomed.

* * *

CHAPTER 20

THE HEAT BALANCE

Thou art weighed in the balances, and art found wanting.
DANIEL, V.27. B.C. 165.

MANY factories are in woeful ignorance of how much steam they are using in various parts of their process ; how much steam they ought to use with their present methods ; how much, or how little, they could use with different methods.

It is quite easy to find out most of these things with a little trouble, a few simple measurements, and some imaginative estimating. As the author believes that the taking out of a heat balance is the most important method of saving steam, or rather is the key to all kinds of steam savings and process improvements, this chapter is the longest in the book.

The investigation of how much heat is being used and the cogitations as to how little might be used are fascinating. Once a factory embarks on the making of a heat balance it is seldom that enthusiasm is not kindled. The making of such a balance generally brings to light so many extravagant processes and practices, that a substantial steam saving is usually obtained forthwith.

558. COSTING. In every factory some kind of costing is done. In the ultra-modern plant the costing system is often so elaborate that it becomes the master instead of being the servant. In some concerns known to the author, the whole business is clogged by a tangle of costing red tape. On the other hand there is the works which can only take out costs by deducting (or adding) its audited profits (or losses) from (or to) its selling price.

The ideal is a simple costing system which will give the maximum of information from a minimum of labour and cause a minimum of irritation. Many technical men are frightened by the idea of costing. Well, don't call it costing. Call it a heat split or a steam balance. It does not matter whether the result is expressed in Btu or in pounds of steam or in shillings. These figures are readily and mutually convertible.

Now costing by itself is not enough, or rather costing is not the first thing to do. Costing can follow the heat balance. It is impossible to cost without first doing some kind of heat use investigation. Costing can be very dangerous if it is used for factory control. The author knows of a concern whose costing system, covering several factories, is very complete. All the steam to each department is metered and the meter readings go to the costing department. This concern has boasted to the author that it can account for all its steam. Actually it can do nothing of the sort. It only knows how much steam it is

using. No account whatever is taken as to how much steam the various processes should use. This concern thinks that by measuring its steam use it must be economical.

The author has come across another distressing effect of elaborate costing control. In one works a small alteration in process technique was proposed in one department and was estimated to give a large steam saving in another department. The proposal slightly increased the costs in the first department in order to secure the large saving in the second. The manager of the first department successfully resisted this improvement for several years because he was afraid that the increased costs of his department would be laid against him.

Before complete costing can be properly done, a heat balance must be taken out. Now a heat balance does not consist only of finding out where the heat goes. It entails finding out how much heat is needed as well, and this is something that the costing department cannot possibly tell us. The taking out of a heat balance is quite an undertaking. Once the balance has been taken out and we realise how technically deplorable we are, the costing office can tell us week by week whether we are improving and at what rate. The heat balance without costing is of very great benefit ; costing greatly enhances the value of the balance. No heat balance that the author has ever seen can by any imaginative stretch encourage any self-satisfaction. Costing is discussed in the next chapter. This chapter deals only with the heat balance.

559. THE TASK AND THE REWARD. To take out a reasonably reliable balance will take many hours of work. It may be that no suitably qualified person has the necessary time. But someone should make the time, because the time could hardly be better spent, and will generally justify an addition to the salary list. The author makes so bold as to assert that 90 per cent. of steam-using factories can save 25 per cent. of their steam in two to three years as a result of taking out a steam or heat balance. Perhaps it is excusable to give the author's own experience.

In 1926 the author's brother made the first attempt. It was not a true balance as the minimum requirements were not included. Steam meters were few and their readings were, perhaps justifiably, mistrusted. Only about half the steam used was accounted for. Little could be done for the next seven years except to carry out desultory steam-saving plant alteration. In 1933 another attempt was made. Much greater care was taken ; a few, more accurate steam meters were available and it was hoped that some 80 per cent. of the steam might be accounted for. The task was completed over the winter half-year of 1933-34, and the steam accounted for added up to 109·9 per cent. This result was so unexpected that it was thought that some gross error had been made, so the performance was repeated, with even greater care, over the next six months. The steam accounted for added up to 93·5 per cent. The total for the whole 12 months was deemed, rightly or wrongly, to mean that practical success had been obtained.

A steam split had been made into every part of the process. This split for the two half-years is given on the opposite page :—

PROCESS STEAM SPLIT

	Winter 1933–34	*Summer* 1934
Melt, tons per week	7,427	9,416
	Average Weekly Steam in lb.	
	Winter	*Summer*
Common to all products—		
Heating syrup for affination	314,044	399,020
Heating melting water	281,673	384,822
Direct steam to melter	511,458	635,625
Carbonatation heating	699,503	918,284
Heating filter press wash water	161,898	332,109
Heating recovery syrups	242,250	359,411
Recovery vacuum pans—1st crop	751,759	909,354
Recovery vacuum pans—2nd crop	432,978	593,462
Recovery vacuum pans—3rd crop	283,968	497,853
Recovery vacuum pans—water drinks	1,949,742	2,471,896
Heating water for liquor char washing	812,287	957,148
	6,442,000	8,459,000
Special to white sugar—		
Vacuum pans	18,202,587	20,628,046
Remelter—direct steam	46,077	52,101
Syrup reheating	884,133	1,093,935
Heating sugar wash water	17,105	34,431
Heating char lights	99,997	109,671
Heating granulator air	104,940	141,779
	19,355,000	22,060,000
Special to yellow sugar—		
Vacuum pans	384,415	370,376
Heating sugar wash water	485	541
Heating syrups	9,119	9,948
Heating char lights	920	898
Evaporating invert	2,228	3,328
	397,000	385,000
Sub-special to packet sugars—		
Steam to remelter	1,556	1,526
Heating remelt water	199	182
	2,000	2,000
Special to golden syrup—		
Heating filter press wash water	35,414	56,946
Vacuum pans—remelts	699,279	481,064
Heating syrups	178,346	126,808
Heating press and char lights	21,299	14,253
Evaporating press and char lights	764,519	585,080
Heating inversion tanks	43,418	30,393
Heating water for syrup char washing	179,447	102,208
Evaporating golden syrup	692,728	537,266
	2,614,000	1,934,000

	Average Weekly Steam in lb. Winter	Summer
Grand total as measured and estimated	28,809,000	32,840,000
Actual steam generated	26,217,000	35,122,000
Steam accounted for	109·9 per cent.	93·5 per cent.
Steam accounted for, weighted average for year	100·5 per cent.	

A serious urge to cut down steam consumption was born, but for many reasons could not be allowed full scope until 1937, when the steam economy drive started in earnest. Here is the result :—

Half year	*Tons of steam-raising coal of standardised heating power per ton of sugar*
Winter, 1936–37	·25
Summer, 1937	·22
Winter, 1937–38	·23
Summer, 1938	·20
Winter, 1938–39	·178
Summer, 1939	·135
Winter, 1939–40	·134

On a process throughput of 8,000 tons a week this shows the following annual coal consumption :—

Before heat campaign $8{,}000 \times 52 \times \cdot 25 =$ 104,000 tons of coal per year
After heat campaign $8{,}000 \times 52 \times \cdot 134 =$ 55,750 tons of coal per year

SAVING = 48,250 tons of coal per year
or 46 per cent

Now it can be said : " This just shows how bad you were." The answer is : " It does indeed ; but most factories are as bad now as we were then."

This spectacular saving was directly impulsed by taking out the first heat balance, however imperfect it may have been. The heat balances before and after the heat saving campaign are shown in Figs. 353 and 354. An exact comparison of Figs. 353 and 354 is not possible because Fig. 353 is based on raw input whereas Fig. 354 is based on refined output. As a result every line in Fig. 354 is about 5 per cent. thicker than it would have been had it had the same basis as Fig. 353.

Fig. 354 is much more complicated than Fig. 353 because much more detailed information was secured. It will be seen that there is now no "Great Unknown", and that the losses from each department are fairly well ascertained.

It may be said that, as the author's factory is large, there is much more scope for large savings. This of course is true. But the saving represented 46 per cent. What does 46 per cent. or even 26 per cent. represent in any

factory ? If the factory is burning only 10 tons a week, a 26 per cent. saving is, at the price most small consumers have to pay for coal, about £750 a year. To what pains will not the management of a small—or large for that matter—factory go to save £750 a year in the shrewd purchasing of supplies or in labour saving plant ? Steam savings are much more satisfactory than most other types of savings. They are seldom dependant, like boiler house savings or purchasing savings, on constant supervision. They are usually secured by the adoption of different methods or alterations to plant. Once done these alterations continue to fulfil their task with little supervision.

The qualities needed by one who is going to take out a heat balance are :—

(*a*) A knowledge of the factory process.

(*b*) Some knowledge of steam (everything needed is in this book).

(*c*) Imagination.

(*d*) The ability to approximate sensibly and even to guess intelligently when exact measurement or calculation cannot yield a result. (There is no use making an elaborate balance to six significant figures if the wetness of the steam at various points is not known. As the wetness must often be guessed, a result to three significant figures is usually all that is worth producing.)

It will probably pay any factory burning 10 to 20 tons of coal a week for steam raising to take on an extra staff, temporarily or permanently, to take out and study heat balances.

As an example of the size of the task and the time required, in Sections 594 to 611 the heat balance of a laundry is discussed. The author spent 3 hours in the laundry and 40 hours writing these Sections and in rough-drawing Figs. 355 and 356. This gives some indication of the time needed by one who has had experience in taking out balances and in drawing Sankey diagrams, and who knows what questions to ask. The author had some smattered knowledge of the laundry processes and of the general standards of performance of laundry plant. This laundry balance is very rough, but it does cover all the essentials and it should at least act as a catalyst to stimulate the expert to do the analysis in the properly detailed manner.

560. ACTUAL AND TARGET. We want to compare the heat we are actually using with the heat that we ought to be using. It is easy to find the amount of heat we are buying ; it is fairly easy to estimate the amount of steam we are making ; it is not so easy to find out how and where this steam is being used. But it can be done with fair accuracy with very few instruments. All this information is, however, of little use until we know how much heat we ought to be using.

561. BOGEY. The actual steam heat that is needed must not be based on pure scientific theory. It must be based on reasonable practical considerations. For example, the electrical equivalent of heat is 3,415 Btu/kWh, but we must not say that a condensing power station ought only to use coal equivalent to $\frac{3{,}415}{12{,}000} = \cdot 285$ lb./kWh. Such a target would be crazy. That is the target

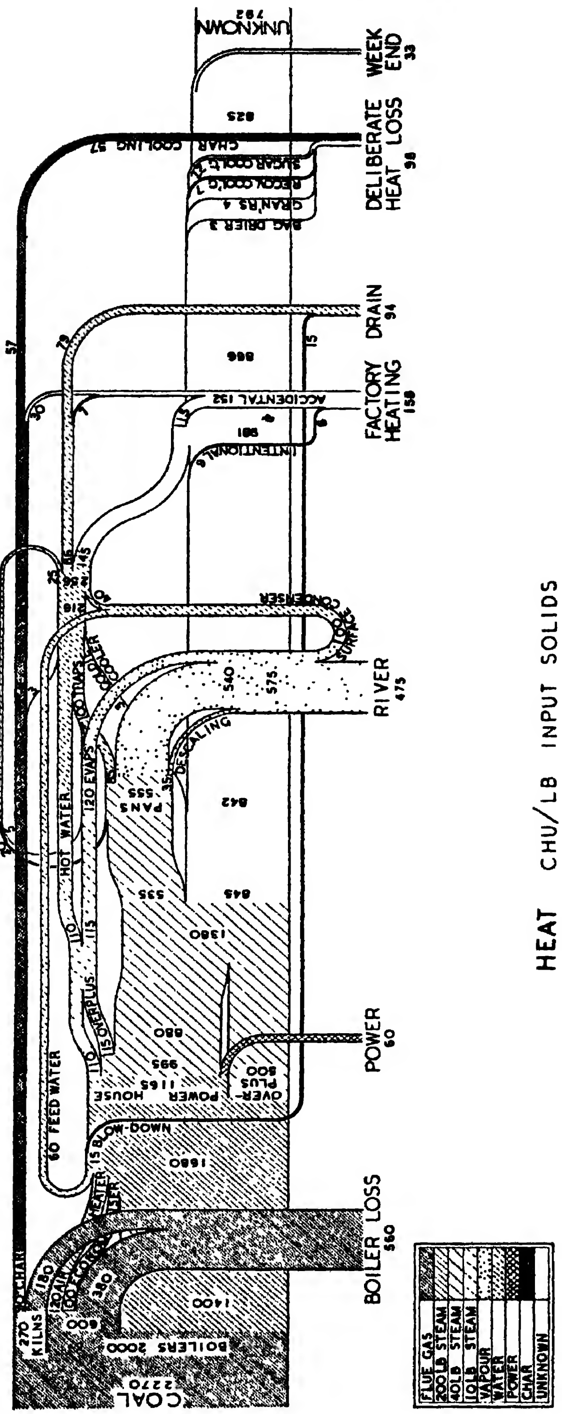

FIG. 353. PLAISTOW WHARF HEAT DISTRIBUTION BEFORE STEAM SAVING CAMPAIGN

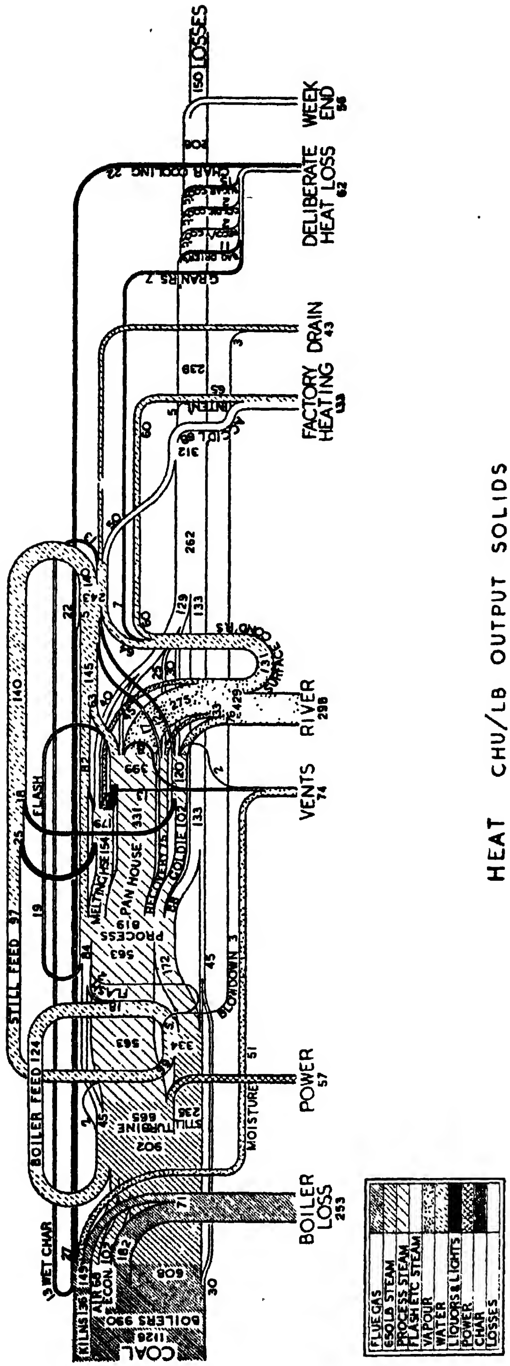

FIG. 354. PLAISTOW WHARF HEAT DISTRIBUTION AFTER STEAM SAVING CAMPAIGN

for a back pressure power plant. On the other hand we should not take the ideal efficiency of the cycle that the station is working nor its best boiler efficiency. The target for a condensing power station is the highest practical boiler efficiency, say 93 per cent., × the highest practical cycle efficiency, say 51 per cent., × the highest reasonable turbine efficiency ratio, say 83 per cent., × the highest possible generator efficiency, say 99 per cent., × (100 − the auxiliaries). It is in the last item that the power stations might do better. Power station designers strive at tremendous cost to squeeze an extra ½ per cent. out of the cycle or boiler efficiency yet seem unmoved by a 5 per cent. consumption for auxiliaries. Reciprocating feed pumps have not received the attention they deserve. Ward-Leonard drive of fans or stokers would seem to have possibilities. Why should not boilers be more often built as vertical towers with air heater and economiser piled on top or arranged as straight horizontal tunnels so that very little mechanical draught would be needed ? In these ways the auxiliaries might be brought down to 3 per cent. The target figure for a condensing power station would then become

$$\frac{3{,}415}{12{,}000 \times \cdot 93 \times \cdot 51 \times \cdot 83 \times \cdot 99 \times \cdot 97} = \cdot 753 \text{ lb. coal/kWh.}$$

Now we have not taken perfection throughout ; that would have meant taking 100 per cent. all round. We have taken "reasonable perfection". One of the best power station performances that the author has heard of is 9,130 Btu/kWh, or with 12,000 Btu. coal, ·761 lb./kWh ; that is 1 per cent. over "reasonable perfection".

Let us take a golfing analogy. The various holes on a golf links are of three general types, long, medium and short. In all short holes it is possible to reach the green in one shot. Occasionally luck will ally itself to skill and the ball will find the hole in one stroke. It would not be reasonable to invoke luck and to say that short holes should always be done in one stroke. But it is reasonable to say that a good player should always reach the green in one stroke and should always hole out in two putts. The good player without luck and without mistakes will always do the short holes in three. Now there are many good players who have occasional luck and who make occasional mistakes so an imaginary player has been invented who never has any luck and who never makes mistakes. His name is "Colonel Bogey" and the number of strokes that he would take is called "Bogey" and clearly means "reasonable perfection". Bogey is attained by many good players and occasionally luck or great skill or both help them to improve on bogey. (James Braid has done short holes at Walton Heath in one stroke on 14 occasions.) Such a player is expected to beat Colonel Bogey and is called a "plus" player. A golfer such as Braid is probably "plus four," which means that in an 18-hole match with Colonel Bogey he would win by four strokes. Bogey is a reference basis for all players good and bad, with which they can compare their performance.

If we can get a "bogey" for our industry or for our particular factory we shall have an invaluable standard with which to compare our performance and a goal for which we can strive.

In an industry producing only a few products that differ little in character the finding of bogey is relatively easy ; but where an assortment of entirely different products are made, bogey is much more difficult to fix. It is, however,

not impossible. Bogey can be based on finished product or on raw material, on weight, bulk or value—tons of raw sugar, pounds of dry laundry, gallons of oil, therms of gas or £100 of product.

While a split of a factory's steam use can be found, and a so-called balance made, by accounting for all the steam generated, this is of little use without some idea as to bogey. Often it is the puzzling over the proper bogey that leads to the major economies. Bogey in a factory or in an industry is not a fixed immutable figure. It must be constantly advanced as technique improves and as better plant becomes available.

Bogey therefore is probably the most important part of the heat balance investigation. The shock received by comparing bogey with actual performance will spur us on to making our balance as detailed and accurate as possible and thus to investigate ways of getting nearer to reasonable perfection.

562. SOME BOGEY EXAMPLES. Bogey in the author's industry is (in 1957) ·1 lb. coal/lb. sugar. In 1937 the coal consumption in the author's factory was ·23 lb./lb. In 1939 it had been reduced to ·135 lb./lb.

In Section 633 bogey for a particular brewery is found to be 12 lb. coal/barrel. The brewery is using 40 lb./barrel and many breweries are using 100 lb. or more.

In Section 611 bogey for a laundry doing a certain type of work is found to be ·54 lb. coal/lb. dry laundry. The laundry thinks it is using 1·3 lb./lb.

Bogey for a particular industry or factory cannot be laid down by someone who has no knowledge at all of the industry or of the factory conditions. For example in Section 486 a beet sugar factory is briefly discussed and it is seen that the obvious bogey figure is some three or four times the coal consumed by many factories. On the other hand a bogey figure worked out by someone who has spent all his life in the industry may be quite unduly generous because old customs and traditions have become unalterable pieces of process technique.

* * *

We will now plunge straight into the welter of heat units, specific heats and whatnot. It is not possible to make a sharp distinction between the finding of bogey and the investigation of heat usage. It is often convenient to look at both simultaneously.

563. INPUT HEAT. All fuel-burning factories, except some collieries and some gas works, know how much fuel they burn because they buy and pay for it and the amount can be found in the ledger. If the heat content of the fuel is known the gross heat input is at once obtained. If the heat content of the fuel is not known it can be guesstimated from Table LXII without going more than about 7 per cent. wrong. (It may be objected that to begin the investigation with a possible error of 7 per cent. is a poor start. This is not so. Nearly every factory that does not know its heat input is using two or more times as much heat as it can properly justify. That is to say it is burning 100 or more per cent. too much. In such cases an error of 7 per cent. is neither here nor there. Those factories like the really good power stations and quite a few industries using only 20 to 50 per cent. more than bogey, know their heat input and can start off with no error.)

564. PRIMARY AND SECONDARY FUELS. Coal and wood are primary fuels. We can rightly take their heat content at its face value. We can possibly do the same with some of the cruder oils, but all the other fuels are secondary fuels which have been derived from primary fuels. For certain purposes, as will be seen shortly, we want to know not only the heat content of the secondary fuels but how much original primary heat content they represent.

TABLE LXII. PROBABLE HEAT CONTENTS OF FUELS

FUEL	HEAT CONTENT BTU — SECONDARY (TECHNICAL)	HEAT CONTENT BTU — PRIMARY
Anthracite	15,000 per lb.	15,000 per lb.
Hard steam coal	14,000 „	14,000 „
Good bituminous coal	13,000 „	13,000 „
Average bituminous coal	12,000 „	12,000 „
Poor bituminous coal	11,000 „	11,000 „
Gas works coke	12,500 „	16,275 „
Coke oven coke	12,500 „	15,650 „
Wood	5,500 „	5,500 „
Petrol	20,000 „	? 30,000 „
Fuel oil	19,000 „	? 20,000 „
Coal tar oil	16,500 „	? 16,500 „
Creosote pitch	16,500 „	? 16,500 „
Coke oven gas	550 per cu. ft.	690* per cu. ft.
Town gas	500 „	650* „
Water gas	290 „	?
Producer gas—hot	160 „	180 „
„ „ —cold	130 „	180 „
Blast furnace gas—cold	100 „	100 „
Electricity—Best condensing stations	3,415 per kWh	10,600† per kWh
„ —Grid average	3,415 „	14,000† „
„ —Grid peaks	3,415 „	30,000† „
„ —Back pressure—large good	3,415 „	5,070 „
„ — „ „ —small good	3,415 „	9,600 „
„ — „ „ —small bad	3,415 „	12,000 „

*† No distribution loss has been allowed for.
* The gas figures should be increased by about 2 per cent. to cover distribution loss.
† These figures should be increased by about 10 per cent. to cover distribution losses.

There are three line of approach when these conversion factors are considered :—

(*a*) The Technical.

(*b*) The Financial.

(*c*) The Fuel Economic.

(*a*) The technical conversion factors are simply the true heat equivalents and these are used where necessary, and are shown in the first column of Table LXII. They are useful for detecting whether the different fuels "as delivered" to the piece of plant are used efficiently or not. It may be very dangerous to use them as a basis for deciding which of two or three fuels should be chosen for a particular process.

(*b*) The financial consideration depends on the heat content per shilling or per pound (£) and their efficiency of utilisation, tempered by the reliability, cleanliness, convenience, capital plant cost, etc., involved.

(*c*) The fuel economic is concerned not only with the calorific value, but the calorific value of the primary fuel that went to the making of the secondary fuel. The efficiency of utilisation must also be considered.

565. ELECTRICITY. The electrical equivalent of heat is 3,415 Btu/kWh, or, roughly, 8,200 kWh per ton of coal. Suppose a patriotic but ill-informed factory management wishes to heat water. It possesses a coal fired boiler with an efficiency of 70 per cent. and is offered an immersion heater with an efficiency of, say, 95 per cent. If the water is to be heated from 60° F. to 200° F., the management would, if they only had the technical heat content in Table LXII, make the following calculation :—

1 ton of coal at 12,500 Btu/lb. liberates 28,000,000 Btu and will heat in the boiler

$$\frac{28{,}000{,}000 \times \cdot 7}{140} = 140{,}000 \text{ lb. of water.}$$

1 ton of coal is equivalent to 8,200 kWh which would heat in the immersion heater

$$\frac{3{,}415 \times 8{,}200 \times \cdot 95}{140} = 190{,}000 \text{ lb. of water.}$$

If only 140,000 lb. of water are to be heated the current needed will be 6,040 kWh which is technically equivalent to ·74 tons of coal.

The factory management therefore orders the immersion heater. What would be the result ? In Chapter 3 the heat consumption of power stations is discussed. The 1957 Grid coal consumption for delivering 1 unit to a customer's premises was 1·4 lb. coal/kWh, or 10,650 × 1·4 = 14,910 Btu. The best base load stations approach 10,000 Btu/kWh and the peaks are taken by stations that burn some 2½ lb. coal/unit or 27,000 Btu/kWh. It is probably fair to say that the load of the factory in question will be taken at average coal consumption when the 6,040 kWh required to heat the 140,000 lb. of water will result in the burning of

$$\frac{6{,}040 \times 1 \cdot 4}{2{,}240} = 3 \cdot 8 \text{ ton coal.}$$

These facts are brought to the notice of the patriotic factory management and they at once cancel their order for the immersion heater.

The managing director lives in London all week but goes on Saturday afternoons to his country bungalow which is fitted with electric fires. With his fine new knowledge he urges the local fuel overseer (it is war time) to allow him to purchase coal so that he can scrap his extravagant electric radiators. Fortunately he announces this proposal to his fellow passengers in the train on Monday morning and one of his companions points out that on Saturday night and Sunday his electric fires will be taken care of by the base load stations, that his utilisation efficiency is 100 per cent. so that to put 3,400 Btu into his room will only cost the Grid 1·1 lb. of coal whereas a coal fire with an efficiency of 20 per cent. will burn 1·36 lb. of coal to supply the same amount of heat. He cancels his request to the fuel overseer.

566. BACK PRESSURE ELECTRICITY. It has been shown in Chapter 3 that by using all the exhaust steam the efficiency of electrical power generation is enormously increased because most of the losses in a turbine go out in the exhaust. The only losses in a back pressure generating plant are friction, leakage to atmosphere, radiation, generator losses and boiler losses. If we take the boiler efficiency of the big plant at 83 per cent. and of the small plant at 70 per cent. ; the generator efficiency as 97 per cent. in big sets and 85 per cent. in small ; and the engine or turbine losses as 7 per cent. in large machines and 25 per cent. in small engines ; if the auxiliaries take 10 per cent. of power in large sets, and 20 per cent. in small sets, we get :—

Electrical Coal Equivalent	*Boiler*	*Machine*	*Generator*	*Auxiliaries*

$$\frac{3{,}415}{12{,}000} \times \left(\frac{100}{83} \text{ to } \frac{100}{70}\right) \times \left(\frac{100}{93} \text{ to } \frac{100}{75}\right) \times \left(\frac{100}{97} \text{ to } \frac{100}{85}\right) \times \left(\frac{100}{90} \text{ to } \frac{100}{80}\right)$$

$$= \cdot 42 \text{ to } \cdot 80 \text{ lb. coal/kWh.}$$

Thus the primary coal equivalent in good back pressure stations lies between 5,070 and 9,600 Btu per kWh. Very bad small back pressure plants may use more than 10,000 Btu/kWh. Each plant must be estimated on its own probable losses for practical heat use investigation, and on its bogey losses for the establishment of bogey.

567. PASS-OUT ELECTRICITY. This is more difficult, especially if the pass-out quantities and the electrical load vary much. The steam to the turbine and the pass-out steam should be metered and averaged over a typical period. The steam quality at the stop valve and at the pass-out point should be ascertained at frequent regular intervals.

Suppose the conditions are :—

Steam to turbine, quantity	100,000 lb./hr.
,, ,, ,, pressure	300 psi.g.
,, ,, ,, temperature	700° F.
,, ,, ,, heat content	1,367 Btu/lb.
Steam at pass-out, quantity	20,000 lb./hr.
,, ,, ,, pressure	40 psi.g.
,, ,, ,, temperature	445° F.
,, ,, ,, heat content	1,257 Btu/lb.

Feed water temperature	.. ..	202° F.
Feed water heat content	.. ..	170 Btu/lb.
Heat put in in boiler, 1,367 − 170	.. .. =	1,197 Btu/lb.
Heat drop to pass-out, 1,367−1,257	.. =	110 Btu/lb.

∴ Heat taken from pass-out steam is equivalent to :—

$$\frac{20{,}000 \times 110}{1{,}197} = 1{,}838 \text{ lb. of high pressure steam.}$$

True steam equivalent used :—

80,000 + 1,838	.. .. .. =	81,838 lb.
Electrical output, say	.. ..	7,500 kW.

Steam heat used :—

$$\frac{81{,}838 \times 1{,}197}{7{,}500} \quad .. \quad .. \quad .. \quad = \quad 13{,}062 \text{ Btu/kWh.}$$

If the boiler efficiency is 81 per cent. this becomes .. 16,126 Btu/kWh gross.

If for some reason good calorimetries of the pass-out steam cannot be got, the problem can be tackled by estimate in similar manner to that described at the end of Section 120 with the aid of Table XIV.

The machine must be treated as two separate turbines, one condensing and one back pressure.

Assume that the adiabatic heat drops are 436 over the condensing machine and 176 for the pass-out machine.

If we refer to Section 104 and Table IX we can estimate the efficiency ratios as 70 per cent. for the condensing machine and 65 per cent. for the pass-out machine.

The output should be :—

$$\frac{(436 \times 80{,}000) \times \cdot 7}{3{,}415} + \frac{(176 \times 20{,}000) \times \cdot 65}{3{,}415}$$

$$= 7{,}150 + 670 \quad .. \quad .. \quad = \quad 7{,}820 \text{ kW}$$

The actual load was .. .. 7,500 kW.

We shall not go far wrong by reducing pro rata when 7,150 + 670 becomes 6,858 + 642.

Although the quantity of steam passed out is small, the machine is quite big and we can probably take the factor of 1·1 from Table XIV.

The heat consumption then is :—

$$\frac{(80{,}000 \times 1{,}197) + (642 \times 3{,}415 \times 1 \cdot 1)}{7{,}500} = 13{,}090 \text{ Btu/kWh.}$$

With a boiler efficiency of 81 per cent. this becomes :—

16,160 Btu/kWh gross.

This agrees remarkably well with the 16,126 Btu obtained in the more accurate manner.

Of course this figure is an overall figure. It can be split between pass-out current and condensing current.

The pass-out figure is :—

$$\frac{3{,}415 \times 1 \cdot 1}{\cdot 81} = 4{,}638 \text{ Btu/kWh generated}$$

The condensing figure is :—

$$\frac{80{,}000 \times 1{,}197}{6{,}858 \times .81} = 17{,}238 \text{ Btu/kWh. generated}$$

These figures must be corrected for the power consumed by the auxiliaries, to get the true heat equivalent of the net available power.

568. TOWN GAS AND GAS WORKS COKE. The conversion factors for electricity, while full of pitfalls, are relatively straightforward compared with gas or coke. Technically, if coal gas contains 500 Btu/cu. ft. and coke contains 12,500 Btu/lb. and if the parent coal contained 12,500 Btu/lb.

1 ton of coal ≡ 56,000 cu. ft. of town gas.

1 ton of coal ≡ 1·0 ton of gas works coke.

When 1 ton of coal is carbonised in a gas works it is converted into saleable secondary fuels in approximately these proportions :—

15,000 cu. ft. gas.

0·5 ton of coke.

As the object of a gas works is the production of gas, it can well be argued that the whole of the conversion losses should be debited to gas, leaving the coke at its primary heat value. Against this it can be argued that both gas and coke are useful secondary fuels and should each bear their share of the losses. Rightly or wrongly, we shall spread the losses over both products.

Technically,

15, 000 cu. ft. of gas	=	·268 ton of coal.
0·5 ton of coke	=	·500 ton of coal.
		·768

As these quantities of secondary fuel were yielded by 1 ton of coal, the coal equivalents must be raised proportionally to add up to unity, when

15,000 cu. ft. of gas	=	·349 ton of primary coal.
0·5 ton of coke	=	·651 ton of primary coal.
		1·000

From these figures we get :—

1 cu. ft. of secondary town gas	=	651 Btu of primary coal.
1 lb. of secondary coke	=	16,275 Btu of primary coal.

569. COKE-OVEN GAS AND METALLURGICAL COKE. Here the object is the production of metallurgical coke ; gas is produced *en passant*, and, as the pillars of pre-war fire by night proclaimed, was sometimes not even a by-product but a waste product. One would therefore be justified in loading all the process losses on to the coke. As coke-oven gas is being more extensively used we will play for safety and load both products equally with the conversion losses.

In a coke oven, some of the gas is used to heat the retort and the result of carbonising 1 ton of coal is approximately

5,000 cu. ft. of gas, and
0·7 ton of coke.

Technically,

5,000 cu. ft. of coke oven gas	=	·0982 ton of coal
0·7 ton of coke	=	·7000 ton of coal
		·7982

Raising these equivalents to a sum of unity gives

5,000 cu. ft. of coke oven gas	=	·123 ton of primary coal
0·7 ton of coke	=	·877 ton of primary coal
		1·000

or	1 cu. ft. coke oven gas	=	690 primary Btu
	1 lb. coke	=	15,660 primary Btu

570. THERMAL VALUE OF OTHER CARBONISATION PRODUCTS. Both gas works and coke oven plants produce benzol and tar. The output of these materials varies greatly but we can take approximately the following for a gas works :—

1 ton of coal yields about

120 lb. tar equivalent technically to	165 lb. of coal
35 lb. benzol equivalent technically to	53 lb. of coal
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Only a small amount of these products are used as industrial fuels. Most of the benzol is used in motor cars and much of the tar is spread on the roads. If we credited coke and gas with the heat contents of these by-products we should get unduly good figures. So that we are not making any very big error by leaving the figures as given in Sections 568 and 569.

571. COMPARISON OF PRIMARY AND SECONDARY FUELS. The comparison may be either technical (as when comparing the actual efficiency of heat use of a particular plant) or financial (the usual factory criterion) or National (when the comparison can only be made in terms of primary heat units). There is a complication with back pressure power whose conversion must be done on a primary fuel basis in order to find the correct financial heat equivalent.

Let us take another simple example. Suppose it is desired to put 1 therm (100,000 Btu) into a room, and we have the choice of using an open fire, a tortoise coke stove, a gas fire or an electric radiator which can use either back pressure current or Grid current. Table LXIII shows the working out and the result from different points of view.

Now this table is provocative, because a certain selection of conditions has been made and a certain line of approach has been taken. The details need not be argued, it is the principle that matters, namely that there are different aspects to the conversion of secondary fuels back to primary which must take account of cost, efficiency of use and, in some cases, the cost to the nation. The most striking thing about the table is that the most economical way of heating a room is by back pressure electricity or coke stove, both from the national and from the personal points of view. This is a very cogent argument in favour of district heating, as it turns the condensing power stations into back pressure stations and any heating that they could not do could be done by coke.

TABLE LXIII. COST TO OWNER AND TO NATION OF HEATING A ROOM

	COAL FIRE	COKE STOVE	GAS FIRE	ELECTRIC RADIATOR	
				AV. GRID	LARGE BACK PRESSURE
Heat to be put in — Btu ...	100,000	100,000	100,000	100,000	100,000
Efficiency of use — per cent. ...	20	60	50	100	100
Btu to oe paid for	500,000	167,000	200,000	100,000	100,000
Cost per unit bought	70*s.*	70*s.*	1*s.*	1·25*d.*	·3*d.*
Cost to heat room	1*s.* 3*d.*	5*d.*	2*s.*	3*s.* 1*d.*	8·8*d.*
Heat units in primary fuel/therm	100,000	130,000	130,000	566,000	148,500
Cost to nation in Btu	500,000	217,000	260,000	566,000	148,500

TABLE LXIV. HEATING A FURNACE

	COAL	PRODUCER GAS	ELECTRICITY	
			GRID	BACK PRESSURE
Thermal efficiency of plant	50	70	90	90
Heat required—Btu	100,000	100,000	100,000	100,000
Heat input—actual	200,000	143,000	111,000	111,000
Primary heat used	200,000	198,000	628,000	165,000

When a factory is deciding what fuel to use or how to get power, it need only consider cost and convenience. But if a heat balance is being taken out the primary fuel may have to be considered. Suppose a heating operation is being done by coal firing in one works, by producer gas firing in another and by electricity in a third. An imaginary comparison is set out in Table LXIV.

If the concern is able to generate its own electricity in sufficient quantity, it would clearly adopt back pressure electricity. Otherwise the question will be settled on the basis of costs alone. But for taking out heat balances and

comparing one method of working with another the primary heat may have to be considered. It may well be that technical considerations are so outstandingly in favour of electricity that all other considerations can be ignored. The same may apply to gas. All that we are here concerned with is that the complications of a secondary fuel must not be overlooked when taking out a heat balance.

This long digression on a rather important matter being finished we can return to bogey and usage.

572. **HEAT CONVERSION.** If the fuel is burnt directly in the process plant, heat input and heat usage are the same ; for example, coal fired ovens, furnaces, retorts and kilns. If the heat is used as steam or producer gas, we must assume some reasonably good efficiency for the conversion of the fuel heat into steam heat or into the secondary fuel—producer gas. This is the figure that we need for bogey. We also want some guide as to the probable efficiency of the actual conversion with various states of plant degradation. We need such a guide for estimating the conversion for our heat balance if we lack the necessary instruments to tell us the painful truth.

573. **STEAM BOILERS.** Table LXV gives the author's views, tempered by those of many more experienced people, of the efficiencies for various types of boilers that we should take for bogey purposes. These figures assume that in all appropriate cases economisers and/or air heaters are fitted. A boiler without an economiser or its equivalent can hardly be entitled to rank as reasonably perfect.

TABLE LXV. "BOGEY" BOILER EFFICIENCIES

	PER CENT.
Crane boilers	50
Shunting loco boilers	60
Vertical cross tube boilers—small	70
„ „ „ „ —large	78
Lancashire boilers	75
Economic boilers	78
Watertube boilers—small	78
„ „ —large	84
„ „ —very large	90
Sectional boilers—small	70
„ „ —large	75

Many factory managements have little idea of how much steam their boilers are generating. It is possible often to do a boiler test. No elaborate instruments are needed. The coal must be weighed and the steam must be measured over a reasonably long period—a week preferably—a shift at least.

If it is impossible to measure the steam, some idea of the efficiency can be got by estimating the losses but, as in the efficiency test, the coal must be weighed. The only instruments needed for estimating the losses are a high temperature thermometer and an Orsat. (The Orsat instrument for determining the CO_2 in flue gas is described on page 720 of the "Efficient Use of Fuel".)

Wherever possible every boiler house should be equipped with such facilities as to give an accurate measurement of the coal burnt each week and the total steam produced.

Measurement of coal burnt should be part of the routine measurements of any factory, however small. If the coal is delivered straight on to the boiler firing floor the measurement can be got with fair accuracy if at the start and finish of a period there is no coal at all on the floor. Estimation of the amount of coal in a heap is difficult, though some experienced men can do it fairly well. The coal can be measured over a shift by putting all the coal fired into a tub frame or barrow, taking care that the tub or frame is always filled to the same amount and weighing a fraction, say one in five or six, of the tubs. It is important that the shift chosen should be a typical average shift and that during the test the firing conditions should be typical average and not greatly improved for the occasion.

The basing of fuel consumption and steam generation on occasional tests is dangerous. It is beyond human frailty to keep the conditions at their average badness during a test and unduly good figures are always obtained. The only safe way is to measure the coal burnt and the steam made in every week.

574. ESTIMATING BOILER EFFICIENCIES. Remarkably accurate guesses of efficiency of a boiler plant can be made by an experienced man. This should never be done by a member of the factory staff. Wishful thinking, however conscientious the thinker, always gives too good a figure. An independent outsider should be asked to make the estimate. Such free expert advice can be obtained from the National Industrial Fuel Efficiency Service or from the Coal Utilisation Joint Council, or—ask a neighbour.

An example of perfectly genuine wishful thinking comes from Scotland. One of the leading firms in the country, who kept good records but who had inadequate boiler house instruments, were logging up week after week boiler house efficiencies of 73 per cent. They were visited by an engineer from the Ministry of Fuel and Power. After a brief look at the boiler house he bluntly said that 73 per cent. was out of the question and that the efficiency was more like 50 per cent. The factory management stoutly maintained that their 73 per cent. was right ; they said it was their test result. On investigation it was found that the test had been done years before and it had been assumed that conditions had remained unchanged ever since. An immediate test was arranged and the efficiency turned out to be 49 per cent. This little true tale points several lessons. Routine logging of figures can be very dangerous. Tests should be done continuously, or, at least, very frequently. An experienced man can guesstimate very exactly.

For what they are worth, Table LXVI gives the author's suggestions as to the most likely boiler efficiencies under various conditions.

575. BOILER TESTS—MEASURING EFFICIENCY. All that we need to do is to measure the coal burnt, the steam put out, the temperature of the feed water and the temperature, pressure and if possible quality of the steam put out. Thermometer pockets must be screwed or welded into the steam pipe and the feed pipe. A recording steam meter should be on every boiler. (The recording and integrating steam meter is the most important instrument in a boiler plant. It tells the fireman his load. It records the peaks and valleys. It records the amount of steam actually made.) If there is no steam meter and if there is no money to buy one, a cheap accurate meter can be made as described in Sections 215 to 232 in Chapter 6, but this meter does not record or integrate so that its readings must be taken every few minutes and the readings logged.

The boiler efficiency is

$$\frac{\text{Pounds of steam produced} \times (\text{total heat of steam/lb.} - \text{heat in feed water/lb.})}{\text{Pounds of coal burnt} \times \text{calorific value/lb. by analysis or from Table LXII}}$$

TABLE LXVI. PROBABLE BOILER EFFICIENCIES

	LANCASHIRE	ECONOMIC	WATERTUBE
No instruments, dirty, brickwork leaking badly, dampers jammed .. .. .. (A)	40	45	50
Few instruments, clean, brickwork leaking, dampers inaccessible .. .. .. (A)	50	55	60
No instruments, clean, brickwork fair, dampers accessible .. .. (B)	60	65	68
Good instruments, all working, tight brickwork, dampers operated from firing floor .. (B)	65	70	73
Deduct 5 if working day work only (A) Add 5 for economiser (B) Add 8 for economiser			

There are refinements and corrections that should be applied such as the measurement of, and correction for the heat in the blowdown. There is however little need to apply any other refinements or corrections while the efficiency is 40 or 50 per cent. As soon as the boiler house has been screwed up to about 65 per cent. we can start doing an accurate test.

It may be that there is a water meter on the feed but no steam meter. The water meter can be used but the measurement of steam is better. Before the water meter readings can mean anything we must know the amount of blow-down. This can be measured, making proper allowance for flash. Water meters, especially on hot water, are not as reliable as steam meters, but if water is to be measured it is better to measure the fill of large feed tanks by dip than to trust to water meter readings. Water meters should be very frequently checked against actual flow into a measured tank. If a continuous record is to be kept based on water to boilers and on blow-down the blow-down cannot be

measured each time. A standard blow-down should be aimed at—say 1 or 2 inches on the glass and this should be always used, the concentration being controlled by the frequency of the blow-down.

For a description of the proper way to carry out an accurate boiler test, see Chapter XVI of " The Efficient Use of Fuel ".

576. BOILER TESTS—MEASURING LOSSES. Many authorities on boiler testing hold the view that it is easier and more accurate to measure the losses than to measure the efficiency, and to deduce the efficiency from the losses. This may be true of a very efficient plant—the author is not qualified to express an opinion—but he stoutly affirms that the measurement, so-called, of losses in a bad plant is impossible and is pure guesswork.

For what it is worth, here is the method. The losses to be measured are

(*a*) Sensible heat in the flue gases.
(*b*) Unburnt carbon monoxide in the flue gases.
(*c*) Combustible matter in ashes.
(*d*) Unburnt coal carried into flues as grits.
(*e*) Radiation.
(*f*) Blowdown.
(*g*) Safety valve blows.

In a good plant with plenty of resources (*a*), (*b*), (*c*) and (*f*) can be ascertained. But however good the plant items (*d*), (*e*) and (*g*) must be guessed or taken at a value based on proper efficiency tests. In a bad plant none of these items can be measured except (*a*).

(*a*) The sensible heat loss in the chimney gases, which is probably the largest single loss in any plant, good or bad, is given by the following expression.

$$\frac{\text{K} \times (\text{Temperature difference between the chimney gas and the air entering boiler plant in °F.})}{\text{per cent. } CO_2 \text{ in chimney gas}}$$

K = ·35 for bituminous coal
K = ·37 for anthracite
K = ·39 for coke

The formula is said to be accurate within 1 per cent. for gas temperatures below 500° F., but with higher temperature such as might be met in boilers without economisers the errors may approach 5 per cent.

(*b*) Unburnt CO is generally very small. It usually only becomes appreciable when a great effort is being made to cut down excess air and to raise the CO_2.

(*c*) With coal with a very high ash and with very bad combustion this loss can be very high. With low ash coals and generous grates it can be very low. It may be anything between 1 per cent. and 15 per cent.

(*d*) With dry fine coal and forced draught this can be quite a big loss. No indication can be given.

(*e*) This can be anything from 2 per cent. to 12 per cent.

(*f*) With bad water and cold feed this can be appreciable, but will never be really large.

(*g*) This may be very large and cannot be properly measured even in a well instrumented plant.

Taking everything into account it is probable that Table LXVI will give a far better result in anything but a very good plant than any attempt to measure the losses. When a boiler is being tested for efficiency and the test can be well and accurately done, it is very useful to check the result by balancing the probable losses against the losses deduced from the efficiency.

577. ECONOMISERS. Quite a good indication of combustion efficiency and/or brickwork tightness can be obtained by measuring the performance of the economiser. One pound of coal requires about 16 lb. of air for good combustion. If the burning of 1 lb. of coal evaporates 8 lb. of water there will be about 2 lb. of flue gas for every 1 lb. of water passing through the economiser. As the specific heat of flue gas is about one quarter that of water, it follows that the heat transferred from the flue gas to the water will result in a 1° rise in water temperature for every 2° drop in gas temperature. If the water and gas temperatures in and out of the economiser are measured and the temperature drop of the gases is twice, or a little more, the temperature rise of the water, then the brickwork is in good condition and combustion is reasonable. If, however, the temperature drop of the gases is less than double the temperature rise of the water, it shows that there is too great a quantity of flue gas, which means that the brickwork is leaking or the combustion is wrong.* If the temperature drop of the gases is much more than twice the temperature rise of the water, it usually means that the economiser setting is leaking air. The condition of the economiser tubes has nothing to do with this measurement, nor has the actual temperature. It does not matter whether the gases lose 2° or 200° provided the water gains 1° or 100°.

578. SUPERHEATER. If a superheater is fitted, some idea of its performance is needed in order to know the heat content of the steam for the heat usage split. Superheaters vary considerably and it is difficult to lay down any rule for the amount of superheat that can be expected. The only real way is to get a suitable high temperature thermometer into the steam pipe and find out the temperature of the steam. If this cannot be done, a rough idea of the superheat can be obtained thus :—

Express the area of the superheater tube surface as a percentage of the boiler heating surface. Then 1 per cent. of extra total heat can be assumed to have been put into the steam for every 5 per cent. of superheater surface in Lancashire boilers and for every 3 per cent. of superheater surface in water-tube boilers. This is admittedly crude but it works out right in a remarkable number of cases.

579. FEED WATER. Feed water temperature can be anything between 50° and 250° F. Neglect to estimate or measure the feed temperature may easily introduce an error of 10 per cent. or more. The best thing is of course

* It is possible for both types of leak to be present together when their effects will counteract giving an apparently good figure of 2. This happening should be noticeable by a reduction in the temperature rise of the water.

to put a thermometer in the feed pipe or feed tank. If no thermometer is available it may be possible to estimate the amount of condensate and the amount of make-up and so to deduce the temperature. It is useful to remember that the hand cannot be held in water at 120° F. but it can be held comfortably against a pipe through which hotter water is flowing.

580. ESTIMATING STEAM PRODUCTION. The amount of steam that is being produced can now be estimated with very fair accuracy, probably within about 5 per cent. Here are two examples :—

Example 1. Dirty, neglected boiler house, where it is necessary to climb over the top of the Lancashire boilers to reach the dampers, only to find that the chain is off the pulley ; no economiser ; no superheater ; feed all cold from well ; burning 40 tons a week of slack to raise steam at 120 psi ; working 47 hours a week.

Heat in coal—say	11,500 Btu/lb.
Heat released per hour	$\frac{11{,}500 \times 40 \times 2{,}240}{47}$ Btu
Total heat in steam—saturated at 120 psi	1,193 Btu/lb.
Heat in feed water (50 − 32) ..	18 Btu/lb.
Heat added in boiler (1,193 − 18) ..	1,175 Btu/lb.
Estimated boiler efficiency (Table LXVI 40 − 5)	35 per cent.
Steam produced per hour	$\frac{11{,}500 \times 40 \times 2{,}240 \times \cdot 35}{47 \times 1{,}175}$
Say	6,500 lb.

Example 2. Fairly clean watertube boiler plant ; draught gauges all working ; no other instruments ; brickwork good ; damper controlled from firing floor ; superheater ; economiser ; pressure 200 psi ; burning 120 tons a week of 140 hours of washed Midland small ; feed water about 50/50 condensate and cold-softened make-up.

Heat in coal—say	12,500 Btu/lb.
Heat released per hour	$\frac{12{,}500 \times 120 \times 2{,}240}{140}$ Btu
Total heat in 200 psi sat. steam ..	1,200 Btu/lb.
Estimated feed temperature	140° F.
Heat in feed (140 − 32)	108 Btu/lb.
Heat added in boiler (1,200 − 108) ..	1,092 Btu/lb.
Area of boiler heating surface ..	1,750 sq. ft.
Area of superheater surface	500 sq. ft.
Superheater as per cent. of boiler ..	29 per cent.
Superheater factor (Section 578) ..	3
Heat addition by superheater (Section 578)	10 per cent.
Heat added in superheater	109 Btu/lb.
Total heat of steam	1,309 Btu/lb.

Total heat put into steam (1,309 — 108)		1,201 Btu/lb.
Estimated boiler efficiency (Table LXVI 70 + 5)		75 per cent.
Steam produced per hour	$\dfrac{12{,}500 \times 120 \times 2{,}240 \times \cdot 75}{140 \times 1{,}201}$	
Say		15,000 lb.

581. GAS PRODUCERS. These vary greatly and it is difficult to lay down guides. Here are some suggestions :—

	Thermal Efficiency per cent. Hot Gas	Cold Gas
Bogey	90	75
Large first class plants	85	68
Fair plant—small	80	60
Bad conditions	65	50

582. HOT WATER BOILERS. The foregoing guides to probable boiler efficiency apply equally to hot water boilers except that, owing to the lower temperatures at which these boilers work, they can sometimes be about 1 or 2 per cent. more efficient than steam boilers of the same size operating under the same conditions, but, as economisers are generally not applicable, a deduction may have to be made.

583. HEAT USAGE—DIRECT COMBUSTION. The total fuel used can be found from the appropriate ledger account, but each separate piece of plant should be measured separately. With coal or coke this is comparatively easy. The fuel fed to each user can be measured in tubs or barrows over a certain number of really typical shifts. A proportion, say 10 or 20 per cent. of the tubs or barrows, must be weighed. This will give quite a good idea as to individual consumption. The individual items should be added together. If they add up to 80 per cent of the total, 25 per cent. can be added to each to make things tally. This is not accuracy, but it is much better than nothing. There is a complication—welcome yet tiresome. The very fact of taking all these measurements will almost certainly produce an appreciable reduction in fuel used, so a watch must be kept for this.

Where gas is used in a large number of furnaces or ovens fed from a common main, matters are much more difficult. Without meters on individual plants or groups of plant it may be necessary to make estimates based on pipe size, valve opening or flame size. This may only give a result within 30 per cent. of the truth, but again this is much better than crass ignorance.

Every possible opportunity should be taken of running one piece of plant alone for a few hours on the main gas meter, when its consumption can be found. This is so much better than guessing, that every effort should be made, on, say, Saturday afternoons or Sunday, to get such solo measurements.

It may be that the preliminary investigation will disclose such a bad state of affairs that it warrants individual metering. It must not be assumed that this means buying dozens of meters ; meters can be hired.

Where producer gas is used estimation is even more difficult. But again, by trying to work one plant at a time and measuring the fuel added to the producer some sort of approximation can usually be made.

584. STEAM USAGE. Broadly speaking there are only two kinds of steam heating plant ; plant where the steam is blown direct into or on to the product being heated ; plant where the steam goes into a heating surface which transmits the heat from the steam to the product and causes the steam to condense inside the steam space. The measurement of steam to engines and turbines has been dealt with in Chapter 7.

Steam meters are of course the best of all ways of measuring steam consumption. If they cannot be made available the condensate can give quite an accurate measurement of steam consumption in heating surface plant, provided corrections are made for flash and for steam quality. This has been dealt with in detail in Chapter 6, Section 234. Where, in the absence of steam meters, the steam is directly injected, the heat consumption must be calculated. This is dealt with in Section 235.

All measurements of steam whether by steam meter, condensate measurement or calculation must be corrected. The steam meter and condensate measurements will include the radiation losses over the plant. The calculated consumption will of course not include such losses.

585. SPACE HEATING. This is a far more important item than is generally realised and it is often overlooked. It is only pertinent in winter ; it becomes non-existent in summer. If a factory had all its plant so beautifully lagged that no heat was lost, it would be necessary to provide a space heating system to make the factory habitable in winter. There is probably no steam-using process factory that is so perfect. Suppose, however, that such a state of affairs could be achieved. The cost of providing the space heating system might deter the management from pursuing a laudable heat saving policy. The solution would be to provide part of the plant in the overcool departments with detachable lagging and so enable the place to be heated in winter with no extra capital cost and no extra heating system.

In nearly every steam using factory loss of heat from the plant provides more than enough heat for warming the place. This inadvertant space heating must be allowed for in bogey, and, as will be seen later, in many process industries, space heating is the biggest single steam user. In Section 633 and Fig. 366 bogey space heating for a brewery is seen to be a little larger than all the brewing process steam requirements.

The heat requirements for space heating can be very accurately estimated. Particulars for making accurate estimates can be found in an admirable booklet issued by the Institution of Heating and Ventilating Engineers called "The Computation of Heat Requirements for Buildings". For a first rough estimate we can get quite a good indication by taking broad simple figures. The requirements vary enormously. The figures about to be given apply to British winter. All the space heating figures become zero in summer, so that autumn and spring must be interpolated.

When estimating the heat required for a building, or extra ventilation requirements in summer, the input of heat from machines, lighting and the occupants must be taken into account.

The heat supplied to a room by its occupants is as follows :—

	Approximate Btu/hr./person
Sedentary work	400
Light manual work	600
Heavy manual work	800

In addition to this heat which is sensible heat there is about an equal amount of latent heat in breath and evaporated perspiration, which increases the humidity of the atmosphere and calls for ventilation.

On balance the extra ventilation requirements nearly cancel the human heating, so that in winter the heat output of the occupants should probably be set against the extra ventilation required. While in summer the sensible heat of the occupants must be got rid of by still more ventilation.

The whole of the current energy used for lighting is dissipated in the building and can be legitimately used as a heating source in winter, but an additional ventilation requirement, if the room is artificially lighted, in summer.

The whole of the energy put into a machine eventually appears as heat. If therefore motors drive machines the total input to the motor is available in winter for heating, and the whole of the heat must be got rid of in summer by ventilation. This power heating contains some pitfalls. If the motor is driving a pump much of the heat energy will not be dissipated in the pump but possibly in another department where the resistance to pumping is being overcome. The same applies to conveyors or elevators. But in the case of machine tools, textile machines and the like, all the power is dissipated in the machine, or in its shafting or drive, or in the material being processed, or in the motor and switchgear.

The electrical equivalent of heat is 1 kWh = 3,415 Btu (see Table VIII).

If a building is found to call for 5 Btu/cu. ft./hr., and has a cubic content of say $50 \times 80 \times 20 = 80{,}000$ cu. ft. it will require 400,000 Btu/hr. to heat it.

If the building contains 20 machines each taking 6 kW the heat dissipated in the machines and motors is $6 \times 20 \times 3{,}415 = 409{,}800$ so that while the machines are running the heating plant would not be required.

This does not mean that heating plant can be dispensed with, except in very exceptional cases. It means that a normal heating plant is required to heat the building up before the start of work, but that when the plant is running the heating can and should be shut down. Only too often the heating plant goes on and all the doors and windows are opened thus wasting all the heat put into the heating plant.

586. VENTILATION REQUIREMENTS. Buildings must be ventilated. The more noxious the process the better must be the ventilation. The greater the number of occupants per unit volume the better must the ventilation be. All the air that is used for ventilation must be heated up to the temperature that

it is desired to maintain. All this good heat goes out with the stale air. Table LXVII gives a rough guide to the heating needs to provide the heat for this ventilating air.

TABLE LXVII. APPROXIMATE HEAT REQUIREMENTS FOR BRITISH WINTER VENTILATION

Ultra-lavish ventilation	4 Btu/cu. ft./hour
Lavish ventilation	3 ,, ,, ,,
Good ventilation	2 ,, ,, ,,
Moderate ventilation	1 ,, ,, ,,
Slight ventilation	·5 ,, ,, ,,

Lavish may mean several things. It may be necessary because there are a great many persons collected closely together, or it may be needed owing to the nature of the process, or it may be that a very high standard is maintained, or it may be due to broken windows and open doors. The last reason must be considered when estimating usage ; the first three for taking out bogey and for estimating usage. This heat is solely that needed to heat the air that is passing through the building.

587. LOSSES FROM FABRIC OF BUILDINGS. Heat passes through the roof, walls and floors of all buildings. The heat loss depends upon the material of which the building is built, the thickness of the roof and walls, the site and exposure of the building. The pamphlet referred to in Section 585 will give much help, but for a first approximation Table LXVIII gives enough information.

Let us assume a well built brick factory with a nearly flat corrugated iron roof with skylights ; ventilation lavish ; dimensions 80 ft. × 50 ft. × 25 ft. :—

Ventilation—	
100,000 cu. ft. at 3 Btu/cu. ft./hr.	300,000 Btu/hr.
Walls (hollow brick)—	
[(80 × 2) + (50 × 2)] × 25 = 6,500 sq. ft. at 8 Btu/sq. ft./hr.	52,000 Btu/hr.
Roof (corrugated iron and glass)—	
3,500 sq. ft. iron at 45 Btu/sq. ft./hr.	157,500 Btu/hr.
500 sq. ft. glass at 36 Btu/sq. ft./hr.	18,000 Btu/hr.
Floor (woodblocks on concrete)—	
4,000 sq. ft. at 5 Btu/sq. ft./hr.	20,000 Btu/hr.
	547,500 Btu/hr.

If the boiler is 75 per cent. efficient, bogey coke consumption would be:—

$$\frac{547{,}500}{\cdot 75 \times 12{,}500} = 58 \text{ lb. coke/hour}$$

It is very noticeable that the chief heat eaters are the roof and the ventilation. Can we therefore reduce the ventilation without lowering our standard

and line the roof with fibreboard ? If the ventilation could be reduced by 33 per cent. and the roof lined (which will cut the roof losses down by three quarters) the heat requirements become :—

	Btu/hr.
Ventilation at 2 Btu/cu. ft./hr.	200,000
Walls	52,000
Roof—iron at 10 Btu/sq. ft./hr.	35,000
glass	18,000
Floor	20,000
	325,000

This reduces the coke consumption to 35 lb./hr.

Whether it is practicable to make such drastic reduction is questionable. For bogey purposes it is probably fair to say that the ventilation should be kept at the high value but that bogey should be taken out on the lined roof, because the roof could have been, or could be lined.

TABLE LXVIII. APPROXIMATE BRITISH WINTER HEAT LOSS FROM BUILDINGS

BTU/SQ. FT./HOUR

WALLS		ROOFS	
Single glass	28	..	36
Double glass	15	..	20
Plain brick, 4½-in.	18	Tiles on battens	45
" " 4½-in. + fibreboard + air space	8	" " boards	30
" " 9-in.	14	" " " and felt	11
" " 14-in.	11		
" " 18-in.	10		
" " 23-in.	8		
For plastered walls .. Deduct	1		
Plastered brick cavity walls, 11-in.	9	Asphalte on 6-in. hollow tiles	15
" " " 16-in.	8	" " " + 1-in. cork	6
" " " 20-in	7		
Stone, 12-in.	15		
" 18-in.	12		
" 24-in.	10		
Concrete, 4-in.	18		
" 4-in. + fibreboard + air space	8		
" 6-in.	16	6-in. + Asphalte	17
" 8-in.	14	" " + 1-in. cork	6
" 10-in.	12	" " + fibreboard shuttering	11
		" " " + air space	8
Corrugated iron	34	..	45
" " + fibreboard + air space	9	..	10
" asbestos	32	..	42
" " + fibreboard + air space	9	..	10
" " on wood tongued and grooved	12		
Plain asbestos	25		
" " + fibreboard + air space	8		
Wood tongued and grooved, 1½-in.	12	+ felt	11

FLOORS

Ventilated wood—bare boards	10
" " —boards + lino or rubber	9
Concrete—bare or granolithic or tiled	6
" + wood block	5

588. COST OF INSULATING BUILDINGS v. HEAT SAVING. The heat insulation of buildings is sadly neglected. Not only is much valuable heat saved, but the heating system can be smaller and cheaper and this saving goes a long way to paying for the cost of insulating the building. Insulated buildings are much more comfortable.

Oscar Faber in 1943 (see Section 805) gave some examples, one of which is summarised below with slight modifications.

Factory building. 300 ft. × 100 ft. × 20 ft. to eaves, say 25 ft. average. Walls and roof corrugated asbestos.

Heat loss per annum with 25° F. temp. diff. ..	3,950,000,000 Btu
Heat loss per annum with insulation ⅝ in. fibre board	1,780,000,000 Btu
Annual saving by insulation	2,170,000,000 Btu

Steam heating system by unit heaters with steam at 40 psi ; flash lost ; condensate returned ; boiler efficiency 65 per cent. Coal 45*s.* per ton.

40 psi steam will give up 920 Btu on condensation. The condensate will flash off 8 per cent. by weight. The feed will consist of 92 per cent. condensate at say 200° F. and 8 per cent. make up at 60° F. The feed temperature will therefore be 189° F. 40 psi steam contains 1,177 Btu/lb. so that the boiler must add 1,177 − 157 = 1,020 Btu/lb.

So that 1,020 Btu are added in the boiler for every 920 that are liberated in heating the factory.

With a boiler efficiency of 65 per cent. 1,570 Btu must be in the coal for every 920 needed by the factory.

The money saved per annum by insulation will be ..	£310	
The cost of the insulation will be, say, 45,000 sq. ft. at 5½*d.* or		£1,030
The saving in the heating plant should be about ..		£400
Net cost of insulation is		£630

Insulation therefore shows a return of £310 on £630 or 49 per cent.

It will be seen that the saving by lagging is so high that it warrants the closest examination.

Here is an example from a different source, a consulting engineer in the Midlands.

During the war he was asked to design a factory of about 1,000,000 cu. ft.

One item in his quotation was " Heat Insulation of Building—£1,600 ".

The client objected to this and demanded its deletion. The engineer replied that he must review the estimate. His new estimate was as follows :—

	£
Extra boiler capacity	600
Extra unit heaters	400
Extra piping and wiring	100
Larger boiler house	200
Extra cost of painting	300
	£1,600
Saving in insulation	£1,600
Extra coal to be burnt	700 tons per year.

So that in this case, FOR NO EXTRA COST AT ALL, some £1,500 a year could be saved.

589. SOCIAL WATER HEATING. A 5 in.* bath requires about 18 gallons. A total immersion bath needs about 32 gallons. A good luxurious hot shower is about equivalent to a 5 in. bath. A wash basin takes rather less than 2 gallons. If the water is at 110° F., and a small allowance is made for heating the bath itself, we can work out the heat requirements which are given in Table LXIX.

The usage of water in canteens and sinks is very difficult to reconcile with requirements. A tremendous waste usually takes place. A basis for bogey canteen needs is given in Table LXIX.

TABLE LXIX. BOGEY SOCIAL WATER HEATING

WASHING	
5-in. bath	12,000 Btu/use
Total immersion bath	20,000 " "
Shower	11,000 " "
Basin	1,200 " "
CANTEEN	
Up to 100 persons (½ gall. at 160° F., ½ gall. at 180° F.)	1,200 Btu/person
100 to 500 persons (⅓ gall. at 160° F., ⅓ gall. at 180° F.)	800 " "
500 to 1,000 persons (¼ gall. at 160° F., ¼ gall. at 180° F.)	600 " "

590. PROCESS HEAT REQUIREMENTS. The two principal uses of process heat are for the heating of water or watery solutions and for the evaporation of water either in evaporators or in driers of some kind. Other heat using processes are the heating of other materials to cause chemical or physical changes, the replacement of heat taken up in endothermic reactions, the distillation of liquids and all kinds of stovings, cookings, etc.

In many of the places where heat is used, it is used to replace heat that is being radiated by the process plant. Should such reheating be included in

* During the war the public were exhorted to limit the depth of water in their baths to 5 inches.

bogey? Probably not. Either the processed material should be kept warm by better lagging or quicker processing, or the reheating, or the original heating, should be done by second-hand heat or waste heat, as is in fact done in many industries.

In order to estimate the bogey requirements in any of these processes it is necessary to know, or at any rate have some kind of idea of, the physical properties of the materials—latent heat, specific heat, etc. For a really detailed investigation the correct figures must be used, but for a preliminary survey it is permissible to make approximations. A few convenient figures are given in Table LXX.

TABLE LXX. SOME PHYSICAL CONSTANTS

	LATENT HEAT BTU/LB.		SPECIFIC HEAT
	FUSION	VAPORISATION	BTU/LB./° F.
Water	144	971	1·0
Copper	75	—	·1
Iron	45	—	·15
Lead	9·5	—	·03
Ammonia	—	540	·5
Carbon dioxide	—	100	·25
Hydrogen	—	—	3·3
Bottle glass	—	—	·15
Many organic solids	30	—	·25
Many organic liquids	—	125	·5
Most ordinary gases	—	—	·25
Most textiles	—	—	·35
Most other dry substances	—	—	·25

Table LXX, particularly the last five items, must be used with great reserve. It is only indicative of the sort of figures that are likely. The actual physical constants must be found and used as soon as possible.

For the ascertainment of bogey it is not necessary to measure the heat supplied to any part of the process (unless the process is a mystery). It is only necessary to know the amount of material in process, the amount of heat it will require to raise it to the process temperature and whether it undergoes an exo- or endothermic reaction. It may be very difficult to put in any kind of meter in many parts of the process. It is often possible to estimate flow of material by deduction; or by stopping the inflow and estimating the outflow; or by stopping the outflow and measuring the inflow by dipping the tank or vessel. The out-put of one department is the input of the next department. Some suggestions for measurement are given in Chapter 6. But, broadly speaking, weight or volume measurements are not needed for the finding of bogey. The measurements are needed when we start trying to bridge the yawning chasm between bogey and performance.

591. CONSTRUCTING THE BALANCE. How is the finished heat balance, either for bogey or for performance, to be set out ? It can either take the form of a financial balance—a formidable list of figures, or it can be a flow sheet diagram, or it can be a Sankey diagram. In the author's opinion, the Sankey diagram is quite the most satisfactory way of setting out the result.

The Sankey* diagram can be used for all kinds of things. It can be used to show the distribution of heat, water, material, costs, losses, etc. It is a tremendous improvement on any form of thin-lined diagram, such as a flow sheet. Every value is instantly impressed on the brain without there being any need to read figures. Chalks or water colours can give an additional set of data which the eye can grasp instantly. Two large examples of Sankey diagrams have already been given in this chapter, and a number of smaller ones were used in Chapter 3.

The drawing of a good Sankey diagram is less a science than an art, which can be acquired with patience and practice. Before the Sankey diagram can be drawn, a row of figures or, preferably, an ordinary flow diagram, must be put down setting out all the data.

592. CONSTRUCTING SANKEY DIAGRAMS. Fig. 36 in Section 116 is a Sankey Diagram in its simplest form. Fig. 34 shows one way of collecting losses together. There is no rule. The real measure of success is whether the presentation is clear to the eye. The object of a Sankey Diagram is to present a perception-hitting picture. The various Sankey Diagrams in Chapter 3 are meant all to be instantly comparable to the eye. Consequently all losses are collected together and turned to the left. This makes Fig. 34, Section 114, a little cumbersome to look at.

In Fig. 353 the main flow of heat travels from left to right with everything that can be accounted for coming out downwards. Fig. 354 shows much heat recirculation and must be incomprehensible to anyone who does not know the process. But this is no disadvantage. The object of a sugar refinery Sankey diagram is to stimulate sugar refiners.

As Fig. 354 is to be compared with Fig. 353, both diagrams must be arranged on the same plan.

It might have been better to have arranged Figs. 353 and 354 so that known useful heat went downwards and known losses went upwards.

593. THE APPROACH TO BOGEY. It is impossible to lay down how bogey is to be ascertained, without some knowledge of the practicable technique of the industry. An estimate of bogey without some inside knowledge may put bogey at some impossibly low figure or at a ridiculously high figure that is beaten by the inefficient. An example of what is meant is given in Section 561 where the obvious bogey for a condensing power station was shown to be utterly unapproachable. The beet sugar factory is an instance at the opposite end.

* Whether Capt. Riall Sankey really invented this type of diagram the author does not know, but Sankey is as good a name as any other and is perhaps an improvement on "fat-line" or "entrail", names that the author and his colleagues used for many years.

The separate heat using processes were set down in Section 486, and amounted to 1,316 Btu/lb. beet, and this might have been taken by the uninformed as bogey. But results using less than 400 Btu have been obtained.

Bogey must be approached with a very open mind. Traditional practice may be bad practice and may be simply tradition and nothing else. Because a dyeworks uses open steam to provide circulation in the dye beck does not mean that this is the only way of providing circulation. The man who has lived his life in an industry gets steeped in its tradition and it may not occur to him that there are some other ways of doing things. That is why it is so good to visit every other kind of factory from which an invitation can be cadged. A visit to any factory, no matter what the industry, always teaches something—it may be only " how not to do it."

Some industrial heat balances will now be discussed in considerable detail. But space forbids that anything other than the main processes in these industries be considered. These small omissions are of no importance, they have no material effect on the main argument. It is hoped that the discussion of these balances will not be too wearisome, because they are of paramount importance. The whole edifice of heat economy is built on the sure foundation of the heat balance, with bogey as the cornerstone. The author has only a rudimentary knowledge of the industries discussed and apologises for any bricks he may inadvertently drop.

In places the data has been modified slightly in order to disguise the actual factory. These modifications have been done in such a way as not to affect the efficiency, or in any way alter the argument.

In no case to be considered has the heat needed for heating up the plant itself been taken into account except in the brewery pasteuriser. In continuously working plant the heat capacity of the plant is usually negligibly small compared to the heat needed by the process. But where the heating operation is done in batches the heat absorbed by the plant may be very important. For example, in steam heated moulding presses the press is often cooled with water after each pressing. The heat capacity of the cured material is often much less than that of the press platens.

In every case the heat needed to warm up the plant should be calculated. The method is given in Section 190 where the heat capacity of a steam pipe was worked out. Any plant can be dealt with in a similar way.

Bogeys have been taken out on the assumption that plant modifications are possible on quite an extensive scale. This does not mean that the factory should forthwith scrap all its plant and re-equip itself. It means that a programme should be prepared so that, as funds become available or plant requires replacement, the new plant can be the right plant. Some savings may be so large, paying for themselves in less than twelve months, as to warrant the immediate scrapping or alteration of quite new plant.

The bogeys ascertained must not be taken as general bogeys for the industries. They are the bogeys for the particular factories. It is impossible to lay down a bogey that is applicable to all laundries or to all breweries. Each factory must ascertain its own bogey appropriate to its own particular type of work.

The factories whose heat balances are about to be investigated must not be taken as typical of either the best, the worst or the average of their industries. They were merely factories whose staffs were kind enough to provide the author with figures. The laundry is below average efficiency, the power station is one of the older medium pressure stations. The purpose of the examples is not to describe the use of steam in the industries but to illustrate ways of attacking the heat balance, and the suggested "improvements" are merely illustrations of what might be done or at any rate what might be thought about.

594. LAUNDRY. The following data is obtained from a laundry :—

Coal. 22 tons per week—definite.

Coal consumption. 1·3 lb./lb. dry laundry—estimate.

Throughput. 38,000 lb. dry laundry per week, say 850 lb. per hour—estimate.

Boiler. Hand-fired Lancashire in fair condition with economiser 120 psi.g. On recent tests said to have given 75 per cent. The general look of things at the time of visit hardly supported this. The efficiency looked as if it might possibly be 65 per cent. or at most 68 per cent.

Steam. Pressure to laundry 100 psi.g.

Power. All produced by slow speed drop valve engine in externally good condition. Average load said to be 115 kW. Load at time of visit 75 kW. Say 100 kW.

Engine exhaust. All to calorifier for water heating. Back pressure unknown. Say 5 psi.g. Thermostat-controlled water temperature at 180° F. by opening exhaust vent to atmosphere—said to be very rare.

Condensate. All collected and returned to boiler feed.

Flash. All lost.

Make-up. Said to be very small. When 10 or 20 per cent. suggested was dismissed as excessive.

Type of work. 12 per cent. wool, 88 per cent. cotton. 5 per cent. of the cotton must be air-dried.

Process. The soiled goods are received, marked and sorted (for degrees and type of dirt, quality and material in peace time—for material only in war time). They are then washed in washing machines with the necessary soap and at the correct temperatures, which are very important. They are rinsed at various temperatures, which are again of great importance, and sent to the hydro-extractors where much of the moisture is removed. They are then dried, after being treated with starch if necessary. The driers are of two main types, contact and air. The contact driers are either presses, roller ironers or calenders. The air driers are either revolving tumblers or chambers in which the goods are hung. Plain cotton goods like sheets are dried on multiroll ironers. Shirts and other things of complex shape are dried on presses which can be partly machines but there is always some hand finishing. Rough towels which cannot be squashed in a roller machine are dried in tumblers. Most woollen goods which also must not be squashed are dried in drying rooms or chambers, as also are shaped cotton goods like coats, etc. The finished goods are sorted back to customer-groups and packed.

595. WASHING AND RINSING. The washing machines were said to be loaded with—

4½ lb. dry cotton/cu. ft.
3¼ lb. dry wool/cu. ft.

This is 20 per cent. heavier than the loading recommended by the British Launderers' Research Association and is presumably purely a war time expedient due to shortage of labour and a desire to economise. It should result in a great saving of heat, but may give poorly washed work.

Typical washing machines are :—

42 in. × 84 in. for cotton,
34 in. × 54 in. for wool.

The load of each machine can be found from tables issued by the B.L.R.A. The washes and rinses are all measured by running dip. The water content cannot be calculated from the dip because the revolving cage and the load pick up an uncalculable amount of water. The B.L.R.A. tables allow for these things and give the actual water content for all running dips and loadings.

The load in washing machines holds back 250 per cent. of its weight of water when the machine is drained.

596. COTTON WASHING.

Machine	42 in. × 84 in.
Loading	4½ lb./cu. ft.
Load	303 lb.
Specific heat of cotton	·35
Water equivalent of cotton load	106 lb.
Water held back by wet load	75 gallons

	DIP	GALLONS OF WATER		TEMPERATURE °F.	HEAT REQUIRED	
		IN MACHINE	ADDED		AS HOT WATER AT 180° F.	AS STEAM
	Inch				Btu	Btu
1st Wash	6	166	166	130	141,280	—
2nd ,,	3	143	68	190	80,900	65,660
3rd ,,	3	143	68	210	83,580	42,340
1st Rinse	9	187	112	180	119,920	—
2nd ,,	10	194	119	140	72,860	—
3rd ,,	15	225	150	110	73,320	—
4th ,,	15	225	150	Cold	—	—
			833		571,860	108,000
						571,860
					Total Heat	679,860

Heat used per lb. cotton	2,244 Btu.
Proportion of cotton in throughput	88 per cent.
Actual cotton washing heat of which 313 Btu is steam heat	1,977 Btu/lb. dry laundry

The dips and the temperatures were obtained from the laundry. The amounts of water in the machine represented by these dips were found in the B.L.R.A. tables. The amount of water added, apart from the first wash, was found by deducting 75 gallons, the water held by the load, from the water in the machine.

The heat required is the heat needed to heat the water from 50° F. to the particular temperature plus the heat needed to heat the load.

Heat needed for first wash is :—

Water 1,660 lb. heated from 50° F. to 130° F.

Load 106 lb. water equivalent heated from 50° F. to 130° F.

Heat needed is $(1{,}660 + 106) \times (130 - 50) = 141{,}280$ Btu.

The hot water supply is at 180° F. so that the 1,660 lb. of water can provide $1{,}660 \times (180 - 50) = 215{,}800$ Btu. This is much more heat than is required so that part of the water must be cold, and no steam will be needed.

The water for the second wash is to be 143 gallons at 190° F.

The heat in the machine must be :—

$$(106 + 1{,}430) \times (190 - 50) = 215{,}040 \text{ Btu.}$$

The heat left in the machine from the first wash is :—

$$(106 + 750) \times (130 - 50) = 68{,}480 \text{ Btu.}$$

Heat required for second wash is :—

$$215{,}040 - 68{,}480 = 146{,}560 \text{ Btu.}$$

The water to be added is 68 gallons which cannot provide this heat, so some steam will be needed.

Let $x =$ lb. of 5 psi steam containing 1,156 Btu/lb. and capable of providing $1{,}156 - 18 = 1{,}138$ Btu/lb.

$$\begin{aligned} \text{Then } 1{,}138x + (680 - x)\ 130 &= 146{,}560 \text{ Btu} \\ 1{,}008x &= 58{,}160 \\ x &= 57{\cdot}7 \text{ lb.} \end{aligned}$$

Check

$$\begin{aligned} 57{\cdot}7 \times 1{,}138 &= 65{,}663 \\ (680 - 57{\cdot}7) \times 130 &= 80{,}899 \\ &\overline{146{,}562} \text{ Btu} \end{aligned}$$

Each wash and rinse must be similarly analysed.

The rinses, being at or below 180° F., call for no steam.

597. WOOL WASHING. The heat and water requirements are found in just the same way, except that at the low temperature at which wool is washed, no steam is needed.

Machine	34 in. × 54 in.
Loading	3¼ lb./cu. ft.
Load	93 lb.
Specific heat of wool	·35
Water equivalent of wool load	33 lb.
Water held by wet load	23 gallons

	DIP	GALLONS OF WATER		TEMPERATURE °F.	HEAT REQUIRED AS HOT WATER AT 180° F. BTU
		IN MACHINE	ADDED		
	Inch				
1st Wash ..	7	68	68	100	35,650
2nd „ ..	8	72	49	100	24,500
1st Rinse ..	10	82	59	100	29,500
2nd „ ..	12	92	69	100	34,500
			245		124,150

Heat used per lb. wool	1,335 Btu.
Proportion of wool in throughput	12 per cent.
Actual wool washing heat	160 Btu/lb. dry laundry.

598. DRYING. Good hydro work leaves about 45 per cent. moisture in the material. We will assume 50 per cent.

Sheets, etc., are dried in calenders or multiple ironers	Contact driers.
Shirts, etc., are dried in presses or are hand finished	
Towels are dried in tumblers	Air driers.
Woollens, etc., are dried in drying rooms or chambers	

Average multiroll ironers use about 3·0 lb. steam/lb. moisture evaporated. Very good performances of below 2·0 lb. have occasionally been reported. We will take 3·0 lb.

Average tumbler steam consumption is about 3¼–3½ lb. steam/lb. moisture. Good low pressure recirculation tumblers can be made to operate with 2·0 lb./lb. We will take 3½ lb. Drying rooms may use more, but we will take 3½ lb./lb.

Much of the work actually put into the driers contains nothing like 50 per cent. of moisture, because, during sorting and waiting, much of the moisture dries off into the atmosphere of the work room. But the latent heat for this drying must have been provided by cooling the work or cooling the air of the room. So for heat balance purposes it is correct and legitimate to assume that the full moisture content as left by the hydro must be evaporated.

	Air Drying	Contact Drying
Wool per cent. of work	12	
Cotton per cent. of work	5	83
	—	—
Material per cent.	17	83
Moisture per cent.	50	50
Moisture to be removed	50% of 17%	50% of 83%
or lb./lb. throughput	·085	·415
Lb. steam/lb. moisture	3·5	3·0
Steam required lb.	·298	1·245
A Total heat in this steam (100 psi.g.)	355 Btu	1,482 Btu
Latent heat given up	263 Btu	1,098 Btu
Heat in condensate	92 Btu	384 Btu
Flash lost (Table IV)	·0395 lb.	·1652 lb.
Flash heat lost	46 Btu	190 Btu
B Heat in recovered condensate ..	46 Btu	194 Btu
Heat used in driers (A — B) ..	309 Btu/lb. dry laundry.	1,288 Btu/lb. dry laundry.

599. **SPACE HEATING.—VENTILATION.** Most of the laundry is more than adequately heated by heat lost by the plant. Only the receiving, sorting and packing rooms require system heating. Were all the plant perfect, it would be necessary to instal a space heating system throughout. So that some of the plant losses are not true losses but must be credited to space heating. So in the following balance winter conditions will be assumed and the heat necessary for space heating and provided by radiation will be credited. As the atmosphere is steamy in the wash house and drying departments the ventilation should probably be on the ultra-lavish scale—see Table LXVII. The ventilation in the sorting, receiving and packing departments need only be good.

The receiving, sorting, packing and delivery rooms are :—

$$60 \times 24 \times 13 = 18{,}720 \text{ cu. ft.}$$
$$60 \times 24 \times 18 = 25{,}920 \text{ ,, ,,}$$
$$28 \times 24 \times 16 = 10{,}750 \text{ ,, ,,}$$
$$52 \times 20 \times 11 = 11{,}440 \text{ ,, ,,}$$
$$66{,}830 \text{ ,, ,,}$$

At 2 Btu/cu. ft./hour this calls for 133,660 Btu/hour or 157 Btu/lb. dry laundry.

The washing and drying departments are :—

$$160 \times 40 \times 11 = 70{,}400 \text{ cu. ft.}$$
$$68 \times 36 \times 13 = 31{,}800 \text{ ,, ,,}$$
$$102{,}200 \text{ ,, ,,}$$

(87664)

At 4 Btu/cu. ft./hour this is 408,800 Btu/hour or 481 Btu/lb. dry laundry. The total ventilation needs are 638 Btu/lb. dry laundry.

600. SPACE HEATING—WALLS. The buildings are all strung end to end on a long narrow site with virtually only two end walls. The walls are of plain 9-in. and 14-in. brickwork.

		9-in.	14-in.
One end wall,	40 × 30 ..	—	1,200 sq. ft.
„ „ „	24 × 18 ..	—	432 „ „
Part end wall,	40 × 12 ..	480	—
„ „ „	16 × 18 ..	—	288 „ „
Side walls,	60 × 15 × 2 ..	—	1,800 „ „
„ „	60 × 11 × 2 ..	—	1,320 „ „
„ „	28 × 12 × 2 ..	—	672 „ „
„ „	52 × 9 × 2 ..	936	—
„ „	160 × 10 × 2 ..	—	3,200 „ „
„ „	68 × 11 × 2 ..	—	1,496 „ „
		1,416	10,408 „ „
Windows about ⅓ wall area ..		472	3,469 „ „
Brick area		944	6,939 „ „
Brickwork loss (Table LXVIII)		14 Btu/sq. ft./hr.	11 Btu/sq. ft./hr.
Window glass loss (Table LXVIII)		28 „ „	28 „ „
Total brick loss		13,216 Btu/hr.	76,329 Btu/hr.
Total window loss		13,216 „	97,132 „
Total wall loss		199,893 Btu/hr. or 236 Btu/lb. dry laundry.	

601. SPACE HEATING—ROOFS. Apart from a very small piece of corrugated iron roof, all the roofs are of slate on boards.

Slate on board—low pitched,	60 × 24 =	1,440 sq. ft.	
„ „ „ „ „	28 × 24 =	672 „ „	
„ „ „ „ „	100 × 40 =	4,000 „ „	
„ „ „ „ „	68 × 36 =	2,448 „ „	
		8,560 „ „	
Add 15 per cent. for pitching			9,844 sq. ft.
Slate on board—high pitched	60 × 24 =	1,440 „ „	
Add 50 per cent. for pitching			2,160 „ „
			12,004 „ „
About 1,000 sq. ft. is glass Leaving slate on board		11,000 „ „	
Corrugated iron—low pitched, lined matchboard (40 × 60) + 15 per cent.		2,760 „ „	

We have therefore (see Table LXVIII)

11,000 sq. ft. at 30 Btu/sq. ft./hr.	..	330,000 Btu/hr.
1,000 sq. ft. at 36 Btu/sq. ft./hr.	..	36,000 ,,
2,760 sq. ft. at 20 (say) Btu/sq. ft./hr.	..	55,200 ,,
		421,200 ,,
	or	496 Btu/lb. dry laundry.

602. SPACE HEATING—FLOORS.

	Wood	*Concrete*	*Wood on concrete*
Dimensions	60 × 24	60 × 24 28 × 24 160 × 40	68 × 36
Area—sq. ft.	1,440	8,512	2,448
Btu/sq. ft./hr. (Table LXVIII) ..	10	6	5
Btu/hour	14,400	51,072	12,240
Total floor loss		77,712 Btu/hr.	
Floor loss		91 Btu/lb. dry laundry	

603. POWER. The engine gets steam at say 100 psi.g. and exhausts at, say, 5 psi.g. A little interpolation in Table X tells us that an ideal engine taking 10,000 lb. steam/hour under these conditions would give 550 H.P. and that the efficiency ratio would probably be 66 per cent. for a practical engine of this size. The laundry engine is giving about 134 H.P. and is therefore about one third of the size.

We can hardly expect to get an efficiency ratio higher than 60 per cent.

The engine in Table X would at 60 per cent. give $550 \times \cdot 6 = 330$ H.P.

The laundry engine gives 134 H.P. and therefore requires

$$\frac{10{,}000 \times 134}{330} = 4{,}060 \text{ lb. steam/hour.}$$

With a dry laundry throughput of 850 lb./hr., this means that the steam to the engine must be $\frac{4{,}060}{850} = 4 \cdot 78$ lb./lb. dry laundry

having a heat content of $(1{,}191 - 18)\ 4 \cdot 78 = 5{,}605$ Btu/lb. dry laundry.

The losses in the engine apart from those that go into the exhaust are, say 20 per cent. so that the heat actually used for power purposes will be

$$\frac{3{,}415 \times 100}{\cdot 8 \times 850} = 502 \text{ Btu/lb. dry laundry.}$$

If the loss is 20 per cent. we get 400 Btu/lb. delivered as power,

102 Btu/lb. generating loss,

and $5{,}605 - 502 = 5{,}103$ Btu/lb. in the exhaust.

604. LAUNDRY HEAT SUMMARY.

Cotton washing	1,977
Wool washing	160
Contact driers	1,288
Air driers	309
Power	502
Space heating—Ventilation	638
Building loss	823
	5,697 Btu/lb. dry laundry.

If the boilers are 68 per cent. efficient with coal of 11,000 Btu/lb. and 1·3 lb. coal/lb. dry laundry, the steam heat produced is

$$11{,}000 \times 1{\cdot}3 \times {\cdot}68 = 9{,}724 \text{ Btu/lb. dry laundry.}$$

The actual heat purchased is 14,300 Btu/lb. dry laundry.

605. LAUNDRY DIAGRAM. We can put all this information down as a Sankey diagram, Fig. 355. This diagram makes no attempt to assess bogey. It is simply a first attempt to put down what the laundry say they are doing and to show what a good picture can be obtained from the somewhat meagre data available.

The exhaust from the engine goes into the calorifier and consists of 4·78 lb. steam containing 5,103 Btu. At 5 psi.g. the latent heat is 961 and the sensible heat is 196. If x is the amount of steam, we have

$$(4{\cdot}78 \times 196) + 961x = 5{,}103$$
$$x = 4{\cdot}33 \text{ lb.}$$

The exhaust must therefore consist of

4·33 lb. steam
·45 lb. moisture

The wash water requires 2,137 Btu which will be provided by exhaust heat up to 180° F. Above that temperature live steam must be used. The live steam heat required was 313 Btu, leaving 1,824 to be provided by engine exhaust.

The amount of exhaust steam condensed in the calorifier will be

$$\frac{1{,}824}{961} = 1{\cdot}9 \text{ lb.}$$

The calorifier trap will have to handle this and the moisture in the steam

$$1{\cdot}9 + {\cdot}45 = 2{\cdot}35 \text{ lb.}$$
$$\text{containing } 2{\cdot}35 \times 196 = 461 \text{ Btu.}$$

The flash from 5 psi steam to atmosphere is 1·58 per cent. (Table IV) or ·0371 lb.

$$\text{containing } {\cdot}0371 \times 1{,}151 = 43 \text{ Btu.}$$

The heat left in the condensate after flashing will be

$$461 - 43 = 418 \text{ Btu.}$$

The steam to the driers and its condensate and flash are all set out in Section 598 and can be drawn straight in.

Only the known condensate has been shown as returning. There is other condensate from the steam that has not been accounted for, but until we can account for it we must leave it out of the diagram unless it can be measured.

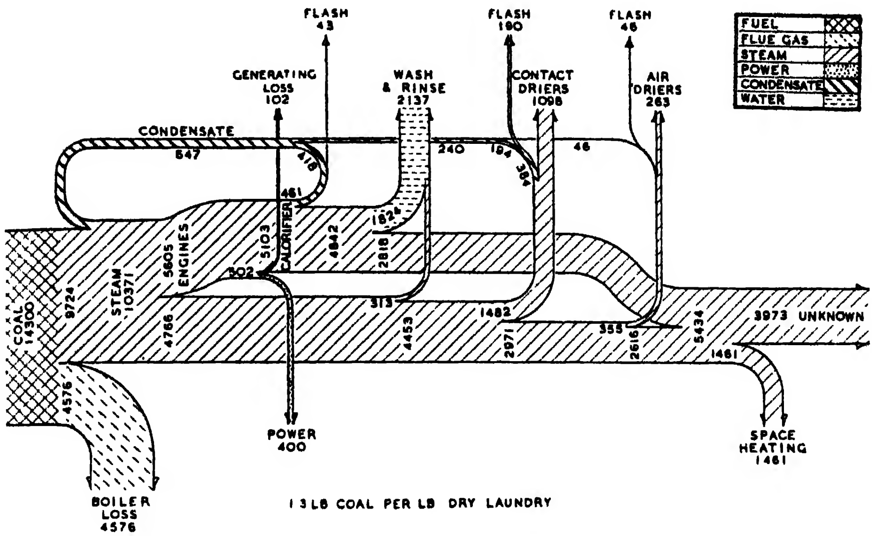


FIG. 355. ACTUAL LAUNDRY HEAT DISTRIBUTION

606. LAUNDRY DISCUSSION. Fig. 355 runs true to form. The author has seldom seen a first industrial Sankey heat diagram that did not look like this. (Compare the first diagram of his own factory, Fig. 353.) It is the striking manner in which the Sankey diagram drives its lessons home that makes it so valuable.

Now this diagram was based largely on what the laundry thought it was doing—not what it was doing. This does not make the picture any better; it might only cause it to be the wrong shape in some places.

Only 45 per cent. of the heat in the exhaust steam has been accounted for and we have taken much less power and consequently have shown much less exhaust steam than the laundry thought they were making. There are some obvious possibilities. Either the exhaust is blowing to atmosphere much oftener than the management believe or there is a tremendous waste of hot water, or the engine is extremely inefficient with very heavy cylinder condensation. Possibly all these things are happening.

The calorifier thermostat is set to heat the hot water to 180° F. This means that much live steam must be used in the washers, thus ensuring that more exhaust steam is wasted.

If the boiler efficiency were really 75 per cent., as believed by the laundry management, it makes the picture much worse as the unknown would go up

from 3,973 to 4,974. If, on the other hand, the boiler efficiency were nearer 50 per cent.—from experience this is by no means unlikely, or unreasonable—over half of the great unknown is accounted for.

The flash steam is all lost. But it only amounts to 7 per cent. of the unknown, and, although it is better to collect than to waste it, the big things should be tackled first. Steam or condensate metering is clearly called for so that the sites of the unknown wastes can be found.

* * *

We will now ascertain, if we can, just how little heat this laundry might use, doing the same work as before, but using the technique laid down as proper by the B.L.R.A.

* * *

607. WASHING BOGEY. We will take washes and temperatures as suggested by the B.L.R.A. for good laundry practice. This will increase the amount of water appreciably. We will also load the washing machines to the recommended loading. This will still further increase heat and water consumption.

We can modify the tabulations in Sections 596 and 597 thus :—

Cotton—

Machine	42 in. × 84 in.
Loading	3½ lb./cu. ft.
Load	235 lb.
Specific heat of cotton	·35
Water equivalent of cotton load	82 lb.
Water held by wet load	59 gallons

	DIP	WATER, GALLONS		TEMPERATURE ° F.	HEAT REQUIRED, BTU		WATER RUN OFF	
		IN MACHINE	ADDED		AS HOT WATER AT 210° F.	AS STEAM	GALLONS	HEAT BTU
	In.							
1st Wash	7	156	156	120	114,940	—	97	67,900
2nd „	4½	142	83	140	88,140	—	83	74,700
3rd „	4	130	71	210	105,640	55,000	71	113,600
1st Rinse	11	187	128	180	146,260	—	128	166,400
2nd „	11	187	128	140	88,320	—	128	115,200
3rd „	17	245	186	110	91,440	—	186	111,600
4th „	20	276	217	Cold	—	—	217	9,540
			969		634,740	55,000		
					55,000			
					689,740 Btu			

Heat used per lb. cotton .. .. 2,935 Btu
Per cent. cotton work 88
Actual cotton washing heat .. .. 2,583 Btu/lb. dry laundry.
of which 206 Btu is live steam heat.

Wool—

Machine .. .. 34 in. × 54 in.
Loading .. .. 2½ lb./cu. ft.
Load .. .. 70 lb.
Specific heat of wool .. .. .. ·35
Water equivalent of wool load .. .. 25 lb.
Water held by wet load .. .. 18 gallons

	DIP	WATER, GALLONS		TEMPERATURE °F.	HEAT REQUIRED, BTU		WATER RUN OFF	
		IN MACHINE	ADDED		AS HOT WATER AT 210° F.	AS STEAM	GALLONS	HEAT BTU
	In.							
1st Wash	13	97	97	100	49,750	—	79	39,500
2nd „	13	97	79	100	39,500	—	79	39,500
1st Rinse	13	97	79	100	39,500	—	79	39,500
2nd „	13	97	79	100	39,500	—	79	39,500
3rd „	13	97	79	100	39,500	—	79	39,500
			413		207,750			

Heat used per lb. wool .. .. .. 2,968 Btu
Per cent. of wool work 12
Actual wool washing heat .. .. 356 Btu/lb. dry laundry.

608. BOGEY WATER HEATING. We have assumed in the foregoing washing tabulation that the calorifier is now arranged to heat the water to 210° F. This has reduced the live steam to the washers.

We will divide the wash house drain into three channels. The hot wash water and first rinse will go into one channel and will pass through a heat exchanger where it will part with some of its heat to the ingoing cold water. The second and third rinses will be recovered and filtered, thus saving this water and the heat in it. The fourth rinse will go straight to drain.

The recovered rinses will be as follows :

	Gallons	*Heat*		
2nd cotton rinse	128	115,200		
3rd „ „	186	111,600		
	314	226,800 for 235	lb. cotton.	
	1·176	849 „	0·88 „	„

	Gallons	Heat		
2nd wool rinse	79	39,500		
3rd „ „	79	39,500		
	158	79,000 for	70	lb. wool.
	·271	135 „	0·12	lb. „
Total	1·447	984		

Assume a heat loss of 10 per cent. and we get
1·447 gallons containing 886 Btu.

The temperature will be $\frac{886}{14\cdot47} + 50 = 111°$ F.

The wash water and first rinses will be :—

	Gallons	Heat		
1st cotton wash	97	67,900		
2nd „ „	83	74,700		
3rd „ „	71	113,600		
1st „ rinse	128	166,400		
	379	422,600 for	235	lb. cotton.
	1·419	1,582 „	0·88	„ „
1st wool wash	79	39,500		
2nd „ „	79	39,500		
1st „ rinse	79	39,500		
	237	118,500 for	70	lb. wool.
	·406	203 „	0·12	„ „
Total	1·825	1,785		

Assume a heat loss of 10 per cent. and we get
1·825 gallons containing 1,606 Btu.

The temperature will be $\frac{1606}{18\cdot25} + 50 = 138°$ F.

The incoming cold water will include the boiler make-up and the last cotton rinse. This will total about 4 gallons/lb. dry laundry, so that there is plenty of capacity to absorb heat.

It is reasonable to assume that half the heat can be recovered, or 803 Btu. This will increase the incoming water by $\frac{803}{40} = 20°$ F. (There are wash house heat exchangers doing better than this.)

The total wash water heat recovery is therefore :—

Recovered rinses	886
Heat exchanged	803
	1,689 Btu/lb. laundry.

609. BOGEY DRYING. For bogey purposes we will take 2 lb. steam/lb. moisture for all driers and we will assume that the moisture is reduced in the

hydros to 45 per cent. We will also assume that only the contact driers require high pressure steam.

We can assume that the flash steam from the drier condensate is used to heat incoming water or that the condensate is heat exchanged.

We can rewrite the drying tabulation from Section 598 thus :—

	Air drying	*Contact drying*
Material per cent.	17	83
Moisture, per cent.	45	45
Moisture to be removed	45% of 17% ·0765 lb.	45% of 83% ·3735 lb.
Lb. steam/lb. moisture	2·0	2·0
Steam required	·153 lb.	·747 lb.
Steam pressure	5 psi	100 psi
Total heat in this steam	177 Btu	890 Btu
Latent heat used	147 Btu	659 Btu
Condensate heat	30 Btu	231 Btu

610. LAUNDRY BOGEY DIAGRAM. Here are the figures we have ascertained from the last three Sections :—

	Btu	*Btu*
Washing heat—Exhaust	2,733	
—High pressure	206	
Wash water heat recovered		1,689
Contact driers—High pressure	890	
Condensate recovered		231
Air driers—Exhaust	177	
Condensate recovered		30
	4,006	1,950
Margin for small processes	500	
Space heating	1,461	
Total heating load	5,967	
Recovered heat	1,950	
Net heating requirements	4,017 Btu/lb. dry laundry.	

We can now build up our diagram, Fig. 356, from the figures we have got out in the foregoing three Sections. A little faking is necessary to make the figures tie up.

	Btu
The total wash water heat is	2,939
The wash water heat recovered is	1,689
Net heat to be supplied is	1,250
The high pressure steam heat is	206
Leaving exhaust heat of	1,044

The exhaust steam needed will be .. .. .. $\frac{1,044}{961} = 1{\cdot}086$ lb.

Containing	1,256 Btu
There will therefore be	212 „
in the condensate and it can be assumed that no flash heat is lost.	
The contact drier steam heat is	890 „
with a condensate heat of	231 „
The air drier heat is	177 „
with a condensate heat of	30 „

The total condensate is	1·086 lb.	containing	212 Btu
	·153 „	„	30 „
	·747 „	„	231 „
	1·986 lb.	„	473 Btu

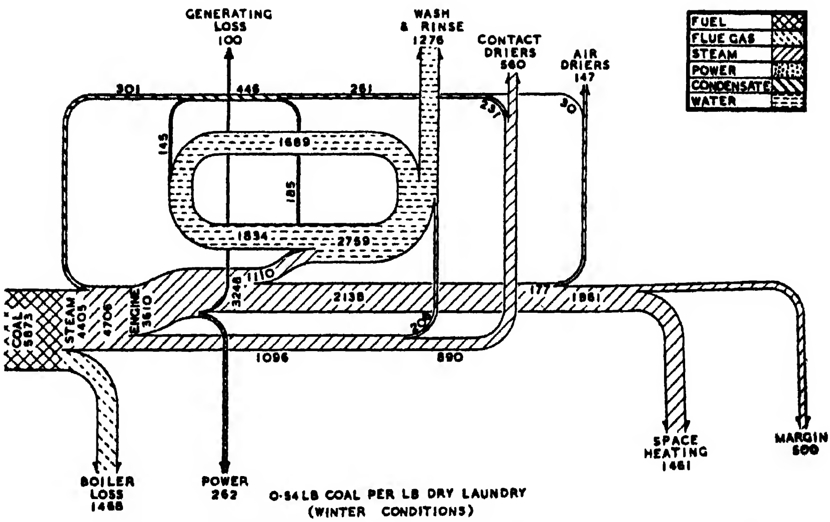


FIG. 356. BOGEY HEAT DISTRIBUTION FOR LAUNDRY ANALYSED IN FIG. 355

The flash steam from the condensate will be used for water heating.

This will reduce the amount of exhaust heat necessary which will in turn reduce the condensate and the surplus condensate heat will itself be reduced. We will make approximate adjustments somewhat arbitrarily as follows :—

Exhaust heat to washing reduced from 1,256 to 1,110.

Condensate from calorifier reduced from 212 to 185.

Surplus condensate heat to water 151 to 145.

611. LAUNDRY POWER. We see by the diagram as it is so far shaped that the exhaust steam is greatly reduced, and that we cannot hope to generate the power that the laundry asserted was their load, with the present laundry engine.

The laundry management believed this load was 115 kW. At the time of the visit it was 75 kW. We took a middle figure of 100 kW.

With 38,000 lb. of work in a 45-hour week the hourly throughput is about 850 lb. 100 kW represents 15·8 H.P./100 lb. work. This is a very heavy power consumption. A fairer figure would be 10 H.P./100 lb.—giving a load of say 65 kW.

For bogey purposes we should take the best engine performance that can be expected.

From Section 104 we find that we can hardly expect a better efficiency ratio than 55 per cent. from a very small engine. If we go to the highest reasonable pressure with a shell type boiler we can say that 250 psi is about the limit and we can superheat to 500° F.

The Mollier chart shows that we can get an adiabatic heat drop of 208 Btu from 250 to 5 psi. With a 55 per cent. efficiency ratio the real heat drop will be 114 Btu, and the exhaust will contain 1,148 Btu/lb. above 32° F. or 1,130 Btu above 50° F.

Our incomplete diagram shows that we have 3,248 Btu in the engine exhaust per lb. of dry laundry. We shall therefore have a power output, with an 80 per cent. generator efficiency, of

$$\frac{3{,}248 \times 114 \times \cdot 8}{1{,}130} = 262 \text{ Btu of power per lb. dry laundry.}$$

This corresponds to just over 10 H.P./100 lb. laundry.

We can take the generator and engine losses at about 100 Btu.

So that the input to the engine must be $3{,}248 + 100 + 262 = 3{,}610$ Btu.

We are justified for bogey purposes in taking a boiler efficiency of 75 per cent., so we can now complete the diagram.

With coal of 11,000 Btu/lb. this laundry should be able to get down to ·54 lb. coal/lb. dry laundry as soon as it can re-arrange and renew its plant, always assuming that the savings are worth it.

It is at present using 22 tons coal/week. If it could achieve bogey it would burn just over 9 tons a week. A saving of some 650 tons a year. At 120*s.*/ton this would be worth £3,900 a year. There is the added advantage that this saving is based on first-rate laundry technique which is far from what the laundry is following at present.

* * *

The laundry is one of the simplest steam using industries. There are only two main processes, the washing and the drying and these are not inter-connected but are entirely independent. We will now look at a more complicated case where the possibilities of heat exchange are much more varied and where there are more processes.

* * *

612. BEER BREWING. The input of a brewery is cold water. The output is cold beer. Apart from losses in spent malt grains, hops and yeast and possibly cask washings, a perfect brewery should need no heat if it had perfect heat exchange and if it could be given the necessary push to start things off.

Fig. 357 shows the information obtained from a brewery. The basic brewery output unit is the barrel of finished beer—36 gallons. All the items are given in terms of this unit. The first process is the leaching of the malt with hot water—or "liquor" as water is called in a brewery. This is done in the mash tun. 35 lb. of malt is used per barrel. The malt is steeped in liquor at about 160° F. for some time. As the "sweet wort" is run off the mash tun additional liquor is added through a sparge pipe at carefully adjusted temperatures until the desired extraction has been obtained. The quality characteristics of beer depend largely upon the analysis of the mashing water and on the temperatures at various stages of mashing.

The "sweet wort" runs into a tank or "under back" until sufficient is collected to charge the copper. (Tanks in breweries are generally called "backs". This word has presumably the same derivation as the dyeworks "beck". Both probably came from the French "bac"—a shallow vessel).

The spent malt, known as brewers' grains, is sold for cattle feeding. 27 gallons of wort are carried away in the grains per "quarter" of malt. (A quarter, which we always thought was 28 lb., proves to be 3 cwt. in a brewery.)

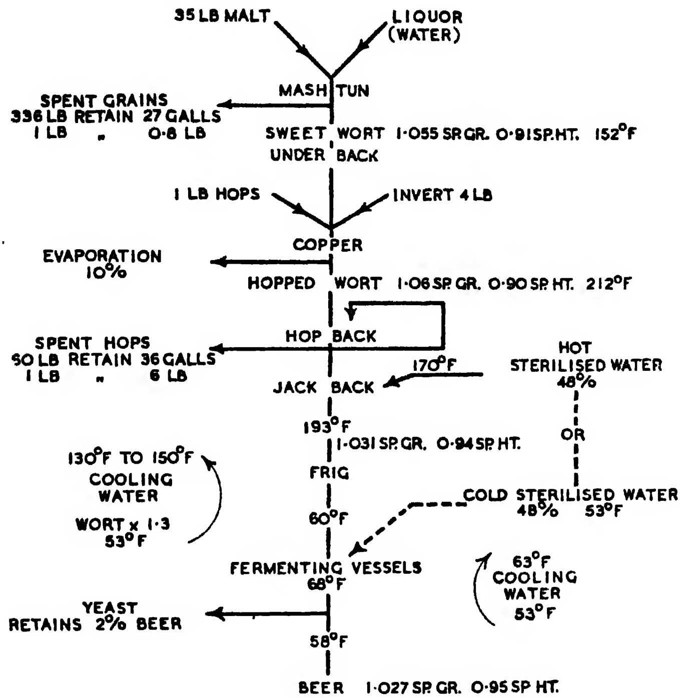
The sweet wort is run into the copper where hops and invert sugar are added. In the copper the wort is boiled for 1 to 3 hours to effect complete sterilisation, to coagulate albumenoids and to extract the flavouring and preservative matters from the hops. Brewers are divided as to the virtue or necessity of evaporation. Some brewers hold that boiling drives off undesirable matters. On the other hand hops are often added later in the process to restore hop oil, etc., driven off in the copper. The object of boiling is not primarily concentration because in many breweries the wort is diluted after the copper. It is maintained by some brewers that pressure cooking with adequate time and circulation will do all that is wanted without any actual evaporation. It might be thought that cutting out evaporation in the copper would bring about a great heat saving, but, as will be seen shortly, provided the vapour heat is being recovered, it would not necessarily effect any saving at all.

From the copper the "hopped wort" is run into a tank called the "hop back" which has a perforated false bottom. It is drawn off the hop back and sprayed back again so that the hops build up a filter bed on the false bottom. The hops filter out the coagulum, hop fragments and other solids, and the hopped wort is circulated through the hop back until it is quite bright. The circulation effects considerable aeration which is often thought beneficial, and by some brewers an essential part of the hop back process. It is then diverted to a storage tank called a "jack back". The reject hops carry away some 5 or 6 lb. of wort per 1 lb. of hops. The hop back and the jack back are unlagged so as to allow some cooling. (This of course is a deliberate waste of heat—see Section 633.)

From the jack back the hopped wort, at about 190° F., is run over the coolers, or "friges" as they are called. These are simply milk coolers on a grand scale. In the brewery under consideration the wort is cooled to 60° F.

and the cooling water, we are told, represents $1 \cdot 3$ times the wort and is heated up to about 130 or 140° F.

If a weak beer is being made, the wort must be suitably diluted. If the dilution liquor is hot sterilised, it must be added before the frig. If it has been cold sterilised, it will be added after the frig. In this brewery the principal product is a mild ale which requires the addition of 48 per cent. of diluting liquor. We shall take out our heat balance with hot and with cold sterilisation. This will give us quite a big surprise.



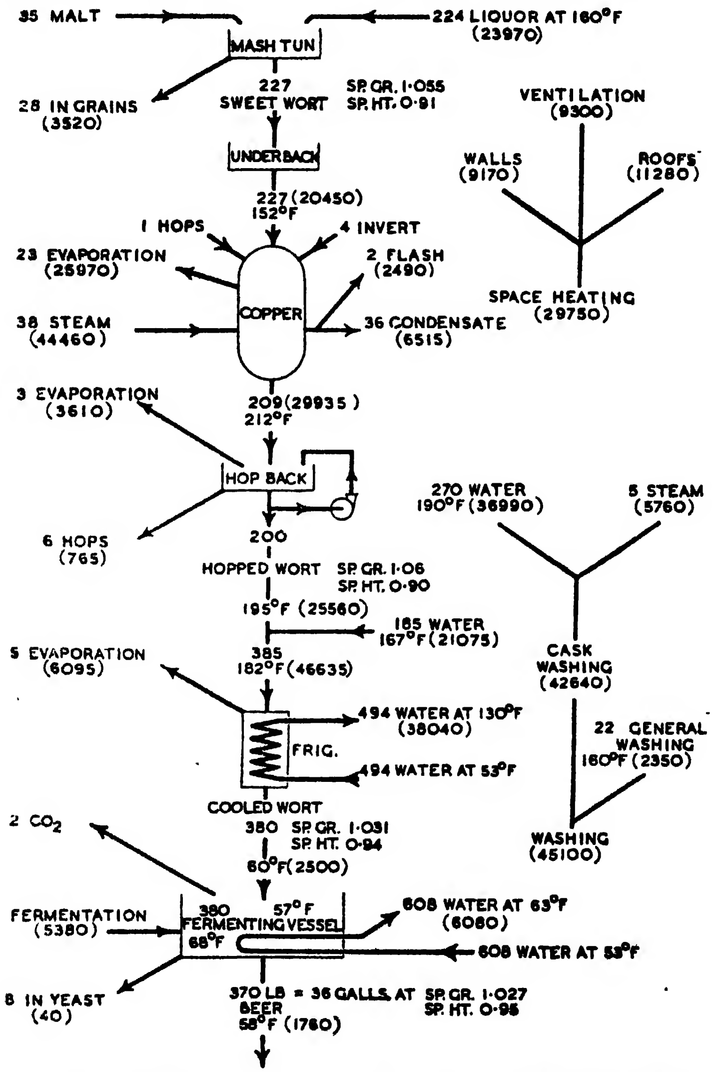
UNIT	36 GALLON BARREL.
CAPACITY	1000 BARRELS PER WEEK.
HOURS	6 A.M. TO 5 P.M. 5 DAY BREWING WEEK. 5 HOUR SATURDAY HEATING.
BUILDINGS	BREW HOUSE – 22 x 22 x 50 FERMENTING ROOMS – 30 x 40 x 15 30 x 45 x 15 OTHERS – 22 x 33 x 20
CASK WASHING	27 GALLS. PER BARREL AT 190°F STEAM: UNKNOWN
BOTTLE TRADE	40% SEE FIG. 359
COAL	49 LB PER BARREL. (? 9 FOR WAGGONS)
POWER	7000 KWH. PER WEEK. (25% ESTIMATED FOR BOTTLE DEPT.)
STEAM PRESSURE	25 TO 30 psi
CONDENSATE	AS FAR AS POSSIBLE ALL RETURNED.
WASHING DOWN WATER	400 GALLS. PER DAY AT 160°F.
HEAT RECOVERY	FRIG WATER TO MASHING. COPPER VAPOUR TO CASK WASHING.

FIG. 357. DATA OBTAINED FROM BREWERY

From the frig. the cooled wort goes to the fermenting vessels, where yeast is added and the wort is fermented into beer. The heat of the reaction would raise the temperature to about 75° F. Cooling, or "attemperating," coils prevent the temperature rising above 68° F. and finally cool the beer down to 58° F.

Certain parts of the brewery need warming and ventilating. This heat would have to be provided separately if there were no heat losses from the plant.

The casks must be washed and steamed. Washing calls for 27 gallons per 36-gallon barrel at 190° F. After washing the casks are steamed. The amount of steam is unknown and must be guessed until it can be measured.



58. BREWING FLOW DIAGRAM. OPEN FIGURES, LB./BARREL. FIGURES IN BRACKETS, BTU/BARREL.

The brewery must be kept spotlessly clean, which calls for much water. Most of the lavish washing down is done with cold water, but a certain amount of hot water is used. This is estimated to be 400 gallons a day at 160° F.

613. THE MAXIMUM ESSENTIAL NEEDS. The thermal starting point of this brewery is the temperature—53° F.—of the well water, and this temperature is taken as zero from a heat point of view.

We will now construct our preliminary heat balance process diagram, in which we will assume no radiation losses and no heat exchangers. Each process will be given the heat it calls for.

Fig. 358 is this diagram and must be built up from the bottom, so that everything can be based on the unit product—the 36 gallon barrel. We are assuming a beer with a specific gravity of 1·027 so that the barrel will weigh 370 lb. avoirdupois. The beer is filled into casks at 58° F. With a specific heat of ·95 it will have a heat content above our arbitrary 53° F. zero of $370 \times \cdot 95 \times 5 = 1{,}760$ Btu. We need only take round figures. In the diagram the heat contents are shown in round brackets ( ).

614. FERMENTING VESSELS. We were told in Fig. 357 that the yeast carried away 2 per cent. of beer. So that 370 lb. represents 98 per cent. of the fermented beer, giving 100 per cent. as 378 lb. There is a loss of CO_2 during fermentation so that a close enough approximation is probably 380 lb. into the fermenting vessel.

The cooled wort with a specific heat of ·94 enters the fermenting vessel at 60° F. It will therefore contain $380 \times \cdot 94 \times 7 = 2{,}500$ Btu. The finished beer is at 58° F. with 1,760 Btu. The heat of fermentation would, we are told, raise the temperature to 75° F. unless this were controlled. The specific heat of the finished beer is ·95, so we will guess the average specific heat during fermentation as ·945. The heat of fermentation must therefore be about $380 \times 15 \times \cdot 945 = 5{,}380$ Btu/barrel.

The yeast carries away 40 Btu, so that the total heat to be removed by the cooling water must be $2{,}500 + 5{,}380 - 1{,}760 - 40 = 6{,}080$ Btu/barrel.

The cooling water enters at 53° F. and leaves at an average of 63° F. The amount of cooling water must therefore be 608 lb.

615. FRIG. We were told that 48 per cent. of adjusting liquor, sterilised at about 170° F., was added to the hopped wort before cooling, and that the hopped wort left the hop back at 193° F. We therefore have

48 parts at $(170 - 53) + 52$ parts at $(193 - 53) \times \cdot 9 = 100$ parts containing 12,168 Btu.

As the mixture will probably have a specific heat of ·94 the temperature will be

$$\frac{12{,}168}{94} + 53 = 182° \text{ F.}$$

We were told that the amount of cooling water used in the Frig. was 1·3 times the wort, that it entered at 53° F. and left at 130° F. or so.

Its weight will be $380 \times 1 \cdot 3 = 494$ lb.

It will carry away $494 \times (130 - 53) = 38{,}040$ Btu. approx.

There is an appreciable evaporation of the wort in passing over the Frig. and this will remove the remainder of the heat.

The input at 182° F. and ·94 Sp. Ht. contains 121 Btu/lb.

So we can equate thus :—

If x is the evaporation in lb.

$$\begin{aligned}
\text{Input hot wort} &= \text{Cooling water} + \text{evaporation} + \text{cooled wort} \\
(380 + x)\,121 &= 38{,}040 + x\,(1{,}150 - 53 + 32) + 2{,}500 \\
45{,}980 + 121x &= 40{,}540 + 1{,}129x \\
5{,}440 &= 1{,}008x \\
5 \cdot 4 &= x
\end{aligned}$$

Input is therefore $380 + 5 \cdot 4 = 385 \cdot 4$, containing 121 Btu/lb. or 46,635 Btu.

Output is :—

Evaporation	..	..	..	6,095 Btu
Cooled wort	..	..	..	2,500 ,,
Cooling water	..	..	..	38,040 ,,
				46,635 Btu

The amount of hopped wort is 52 per cent. of the adjusted wort.

52 per cent. of 385 is 200, which, at 195° F. with ·9 specific heat will contain 25,560 Btu.

The adjusting liquor will be 185 lb. containing $46{,}635 - 25{,}560 = 21{,}075$ Btu and will have a temperature of 167° F.

616. HOP BACK. We know from the information in Fig. 357 that each barrel is treated with 1 lb. of hops and that about 6 lb. of thin wort are retained by the spent hops left behind in the hop back. So that there must have been 206 lb. of wort in the hop back at clarity point.

A considerable amount of steam is given off by the recirculation of the wort over the hop bed, which is the principal reason for the drop in temperature over the hop back—from 212° F. to 195° F.

We can equate thus :—

If x is the amount of evaporation

$$\begin{aligned}
(206 + x) \times (212 - 53) \times \cdot 9 &= \left(\frac{206}{200} \times 25{,}560\right) + 1{,}129x \\
29{,}480 + 143x &= 26{,}325 + 1{,}129x \\
3{,}155 &= 987x \\
3 \cdot 2 &= x
\end{aligned}$$

3·2 lb. of evaporation will carry away say	..	3,610 Btu
The bright hopped wort contains ..	..	..	26,325 ,,
Hopped wort leaving copper contains	..	..	29,935 Btu

The spent hops retain 6 lb. of wort and will carry away

26,325 − 25,560 = 765 Btu.

Check :

The output of the copper must be 200 + 6 + 3·2 = 209·2 at 212° F. Sp. Ht. ·9 containing .. .. 29,935 Btu.

617. COPPER. There is a 10 per cent. contraction by evaporation in the copper.

So that 209 lb. out is 90 per cent. of the charge which must have been 232 lb.

But 1 lb. of hops and 4 lb. of invert sugar were added so that the sweet wort must have been 227 lb.

The input sweet wort is at 152° F. with a specific heat of say ·91.

The sweet wort heat content must have been 20,450 Btu.

The evaporation must have been 232 − 209 = 23 lb.

Heat carried away by evaporation is 23 × 1,129 = 25,970 Btu.

We therefore have :—

	Btu
Vapour off	25,970
Hopped wort out	29,935
	55,905
Sweet wort in	20,450
Steam heat needed	35,455

The latent heat of 25 psi steam is 935 Btu, so that $\frac{35{,}455}{935}$ = 38 lb. of steam will be required.

This will contain 44,460 Btu of total heat.

The flash is lost, and Table IV shows that it will be 5·7 per cent. There will therefore be a flash loss of 38 × ·057 = 2·166 lb. containing

2·166 × 1,149 = 2,490 Btu.

The heat in the condensate will be

44,460 − 2,490 − 35,455 = 6,515 Btu.

618. MASH TUN. The output from the mash tun is 227 containing 20,450 Btu.

Fig. 357 tells us that 35 lb. of malt will carry away 28 lb. of thin wort, practically water.

The sweet wort has a specific gravity of 1·055. The solids in solution are chiefly sugars or the like so we can assume a solids content of about 13·5 per cent.

The 227 lb. of sweet wort will therefore be made up of 196 lb. of water and 31 lb. of solids.

The liquor input to the mash tun must be $196 + 28 = 224$ lb.

The average temperature of the mashing liquor is 160° F.

The heat required by the mashing liquor is

$$224 \times (160 - 53) = 23{,}970 \text{ Btu.}$$

The sensible heat for heating the malt can be ignored because mashing is exothermic and provides some heating.

The spent grains carry away 28 lb. of thin wort. We can approximate by saying that the wet grains carry away the difference between the input liquor heat and the outgoing sweet wort heat.

$23{,}970 - 20{,}450 = 3{,}520$ Btu carried away in the wet grains.

619. SPACE HEATING. The brewery is steamy and calls for good ventilation. We will therefore, from Section 586, say that we need 2 Btu/cu. ft./hr. for heating the brewery ventilating air.

The cubic content of the brewery is $22 \times 22 \times 50 = 24{,}200$ cu. ft. or 48,400 Btu/hr.

The fermenting rooms require good ventilation to clear CO_2.

Their cubic content is $30 \times 85 \times 15 = 38{,}250$ cu. ft. or 76,500 Btu/hr.

The other small departments require only moderate ventilation and have a content of $22 \times 33 \times 20 = 14{,}520$ cu. ft. or 14,520 Btu/hr.

The losses from building fabrics are given approximately in Table LXVIII, Section 587.

The stone walls are

$$\begin{aligned}
22 \times 50 \times 4 &= 4{,}400\\
85 \times 15 \times 2 &= 2{,}550\\
30 \times 15 \times 2 &= 900\\
33 \times 20 \times 2 &= 1{,}320\\
\\
&\quad 9{,}170 \text{ sq. ft.}
\end{aligned}$$

At 15 Btu/sq. ft./hr. the loss from the walls is about 137,550 Btu/hr.

Tiled roof

$$\begin{aligned}
22 \times 22 &= 484\\
85 \times 30 &= 2{,}550\\
22 \times 33 &= 726\\
\\
&\quad 3{,}760 \text{ sq. ft.}
\end{aligned}$$

At 45 Btu/sq. ft./hr. the roof loss is about 169,200 Btu/hr.

The brewing hours are from 6.0 a.m. to 5.0 p.m. for five days ; and from 7.0 a.m. to 12.0 noon on Saturdays heating will be needed. We can assume

that heating is required for about 67 hours a week. The output is 1,000 barrels per week, or 15 barrels per "heating hour". The space heating load is therefore—

		Per hour	Per barrel
Ventilation :	Brew house	48,400	3,230
	Fermenting	76,500	5,100
	Others	14,520	970
Fabric :	Walls	137,550	9,170
	Roofs	169,200	11,280
			29,750 Btu/barrel.

It has here been assumed that it is necessary to heat the ventilating air in the fermenting rooms. This may not be necessary, in which case the space heating requirements would be reduced by 17 per cent.

620. WASHING HOT WATER. We know from Fig. 357 that the cask washing water is about 27 gallons per barrel at 190° F.

This requires a heat of $270 \times (190 - 53) = 36{,}990$ Btu/barrel.

There is in addition an unknown amount of steam used for sterilising the casks. We will guess this as being 5 lb./barrel until it can be measured and the requirements found out.

Steaming heat is $5 \times (1{,}173 - 53 + 32) = 5{,}760$ Btu/barrel.

General washing uses 400 gallons per day at 160° F. Call this 2,200 gallons per week or 2·2 gallons per barrel containing $22 \times (160 - 53) = 2{,}350$ Btu/barrel.

621. BOTTLING—REFRIGERATION. We were told that 40 per cent. of the trade is done in bottles, or 400 barrels per week and we have the information given in Fig. 359. From this we can construct the process diagram Fig. 360.

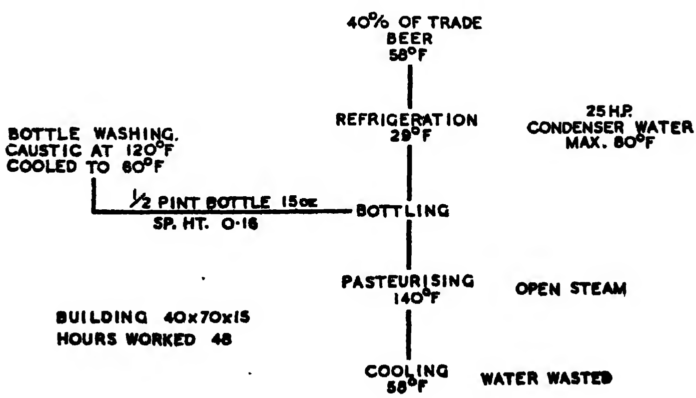


FIG. 359. BOTTLING DATA OBTAINED FROM BREWERY

The beer leaves the brewery at 58° F. and is cooled by refrigeration to 29° F.

The heat to be removed is $370 \times (58 - 29) \times \cdot 95 = 10{,}200$ Btu/barrel.

The refrigerator is driven by a 25 H.P. motor. Assume that the refrigerator runs for 50 hours a week at an average of 20 H.P., then the extra mechanical heat to be removed is

$$\frac{20 \times 2{,}545 \times 50}{400} = 6{,}360 \text{ Btu/barrel.}$$

The total heat to be removed by the refrigerator condenser is

$$10{,}200 + 6{,}360 = 16{,}560 \text{ Btu/barrel.}$$

If the cooling water enters the condenser at 53° F. and leaves at 80° F. we shall need

$$\frac{16{,}560}{(80 - 53)} = 613 \text{ lb. water per barrel.}$$

The cooled beer now contains $1{,}760 - 10{,}200 = -8{,}440$ Btu/barrel. (This is a minus quantity because our basic temperature is 53° F.)

During cooling the beer clarifies and absorbs CO_2.

622. BOTTLE WASHING. The bottles are washed with a caustic solution at 120° F.

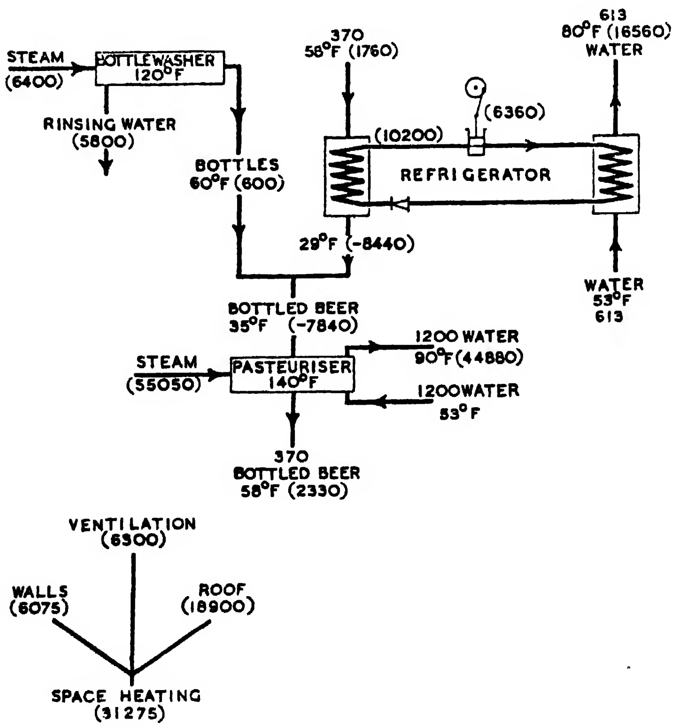


FIG. 360. BOTTLING FLOW DIAGRAM PER BARREL BOTTLED

To bottle one barrel requires $36 \times 8 \times 2 = 576$ half-pint bottles.

These weigh $\frac{576 \times 15}{16} = 540$ lb. with a specific heat of about ·16.

The heat in the bottles after washing is

$$540 \times (120 - 53) \times \cdot 16 = 5{,}800 \text{ Btu/barrel.}$$

The amount of make up heat to maintain the caustic at 120° F. due to loss of caustic and radiation is not known, so we will estimate it at 10 per cent. of the heat needed to heat the bottles.

This gives a total heating input to heat the bottles of say 6,400 Btu/barrel.

The bottles are rinsed free of caustic and cooled to about 60° F. with water which must be wasted.

The cooled bottles contain $540 \times (60 - 53) \times \cdot 16 =$ say 600 Btu/barrel.

The bottle rinsing water carries away $6{,}400 - 600 = 5{,}800$ Btu/barrel.

623. BOTTLING. 370 lb. of beer at 29° F. with a specific heat of ·95 are bottled in 540 lb. of bottles at 60° F. with a specific heat of ·16. So that the bottled temperature is

$$\frac{(370 \times 29 \times \cdot 95) + (540 \times 60 \times \cdot 16)}{(370 \times \cdot 95) + (540 \times \cdot 16)} = 35° \text{ F.}$$

624 PASTEURISING. After bottling, the bottled beer is pasteurised in a steamer at 140° F.

The heat to be added is

$370 \times (140 - 35) \times \cdot 95 =$	37,000	Btu to the beer.
and $540 \times (140 - 35) \times \cdot 16 =$	9,070	Btu to the bottles.
	46,070	Btu
Correction from next section	8,980	,,
	55,050	Btu/barrel bottled.

625. COOLING. After pasteurising, cooling water is sprayed over the bottles to cool them in the pasteuriser until they are cooled to about 58° F.

This calls for the dissipation of :—

$370 \times (140 - 58) \times \cdot 95 =$	28,820	Btu from the beer.
$540 \times (140 - 58) \times \cdot 16 =$	7,080	Btu from the bottles.
	35,900	Btu/barrel bottled.
In addition the pasteuriser is cooled, requiring, say 25 per cent. more cooling or an additional ..	8,980	,, ,, ,,
Total cooling in the pasteuriser ..	44,880	Btu/barrel bottled.

If the water enters at 53° F. and leaves at an average of 90° F. we shall need 1,200 lb. of water which must be wasted.

The heat removed from the pasteuriser during cooling must be put back in the next steaming and this calls for an extra 8,980 Btu which is the correction added to the heats in the foregoing section.

626. BOTTLING DEPT. SPACE HEATING. We shall guess that the ventilation need only be moderate.

The ventilation heat requirements will be $40 \times 70 \times 15 = 42{,}000$ cu. ft. or Btu/hr.

As the bottling dept. abuts the fermenting room there are only three walls to be considered.

These walls have an area of $(70 \times 15 \times 2) + (40 \times 15 \times 1) = 2{,}700$ sq. ft.

At 15 Btu/sq. ft. the total loss will be 40,500 Btu/hr.

At 45 Btu/sq. ft. the roof loss will be $70 \times 40 \times 45 = 126{,}000$ Btu/hr.

The total space heating requirements will be

	Btu/hr.
Ventilation	42,000
Walls	40,500
Roof	126,000
	208,500

If the heating is needed for 60 hours a week for a bottled output of 400 barrels per week the heating requirements for the bottling dept. will be

$$\frac{208{,}500 \times 60}{400} = 31{,}275 \text{ Btu/barrel bottled.}$$

627. BOTTLING HEAT REQUIREMENTS. We must convert all the bottling heat requirements from terms of bottled barrels to total barrels by taking 40 per cent. of each item thus :—

		Btu per barrel bottled		*Btu per barrel brewed*	
Refrigeration—	Heat removed	10,200		4,080	
	Power input ..	6,360		2,540	
	Heat to cooling water ..	16,560		6,620	
Bottle washing	.. ..		6,400		2,560
Pasteurising	.. .. ..		55,050		22,020
	Cooling ..	44,880		17,950	
Space heating	.. ..		31,275		12,510
					37,090 Btu/barrel brewed.

The bottling department is shown in terms of brewed barrels in Fig. 361.

628. TOTAL MAXIMUM HEAT REQUIREMENTS. We can take the items from Figs. 358 and 361 and build up the Sankey diagram, Fig. 362.

Mashing liquor	23,970
Copper (+ flash loss)	37,945
Adjusting liquor	21,075
Space heating brewery	29,750
Cask washing	36,990
Cask steaming	5,760
General washing	2,350
Bottle washing	2,560
Pasteurising	22,020
Space heating bottle department..	12,510
	194,930 Btu/barrel brewed

If we take the not very ambitious figure of 65 per cent. for reasonable boiler efficiency, the total maximum heat input to the brewery should be

$$\frac{194{,}930}{\cdot 65} = 299{,}900 \text{ Btu in the coal.}$$

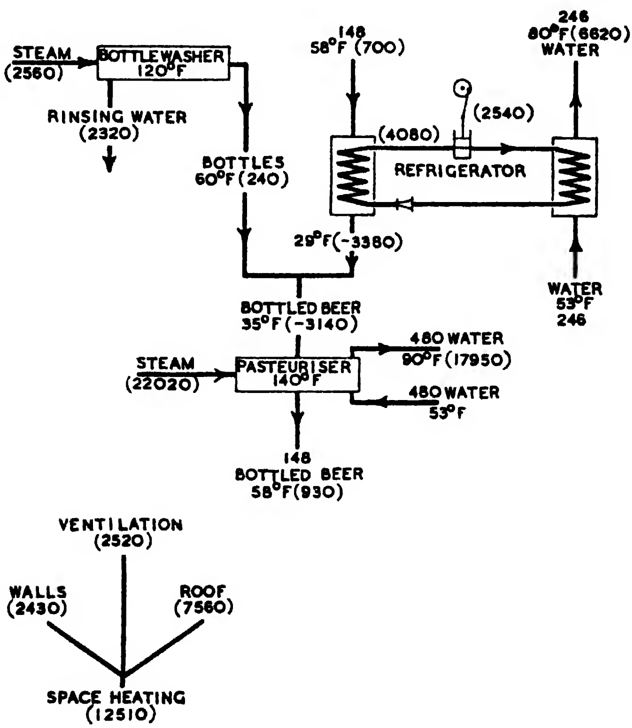


FIG. 361. BOTTLING FLOW DIAGRAM PER BARREL BREWED WITH 40 PER CENT. BOTTLED

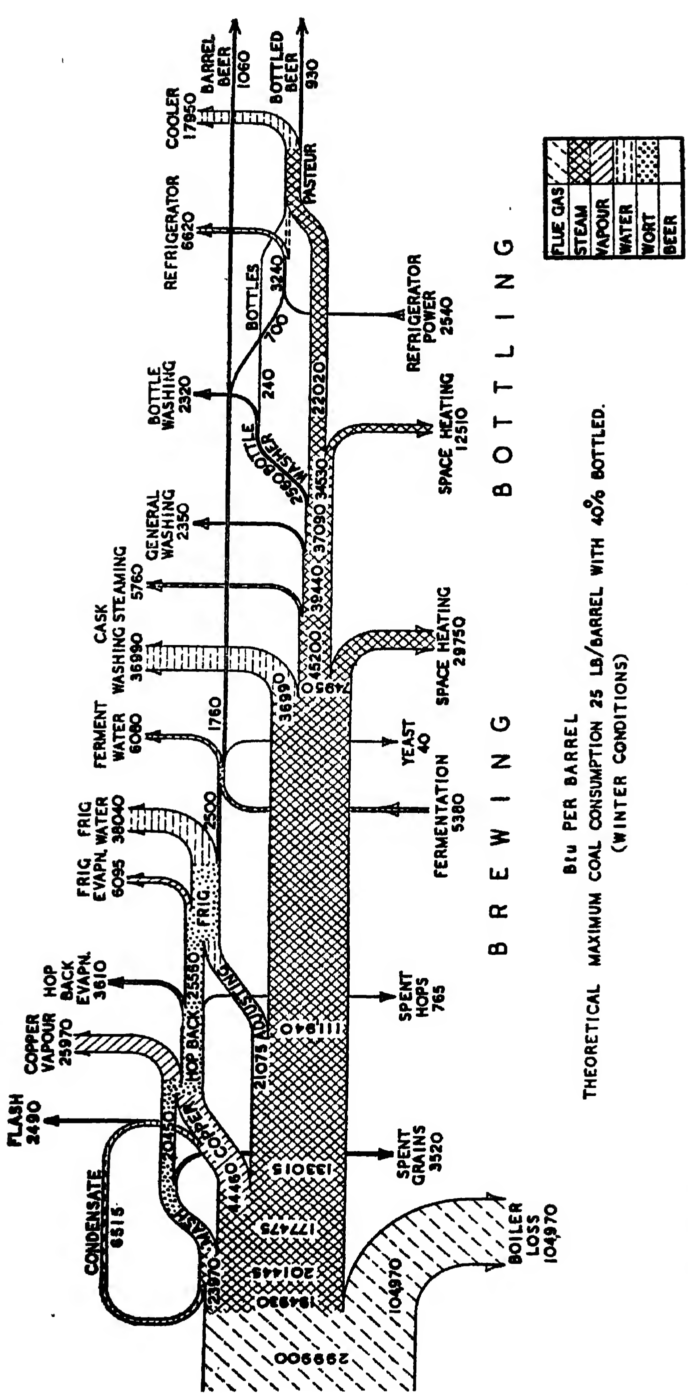


FIG. 362. MAXIMUM THEORETICAL HEAT REQUIREMENTS OF BREWERY WITHOUT HEAT RECOVERY WITH 40 PER CENT. BOTTLED

If we take the coal at 12,000 Btu/lb. the coal consumption should be 25 lb./barrel.

Fig. 357 tells us that the brewery was using 40 lb./barrel.

Fig. 362 shows these maximum heat requirements in the inimitable way that nothing but a Sankey diagram can. While we have called the heat requirements "Maximum", they include no radiation losses other than those needed for space heating. There are no provisions for reprocessing, etc. They omit certain small steam users—sugar-dissolving, etc.

But the picture shows the essential processes done in the most extravagant way with no attempt at heat recovery. In spite of the fact that the brewery in question does do quite a lot of heat recovery, it is using 60 per cent. more coal than can be accounted for.

629. HEAT RECOVERY. We will now investigate the conditions when doing the greatest possible amount of heat recovery, using the same processes as are shown in Figs. 358, 361 and 362.

Let us write down our low temperature requirements and our possible sources of recoverable heat.

	Lb.	*Temp.*	*Heat*	
Requirements :		*°F*	*Btu*	
Mash liquor	224	160	23,970	Max. 180° F., Min. 155° F.
Adjusting liquor	185	167	21,075	
Cask washing	270	190	36,990	
General washing	22	160	2,350	
	701		84,385	
Waste heat sources :				
Copper vapour	23	212	25,970	
Frig. water	494	130	38,040	Min. below 130° F.
Fermentation water ..	608	63	6,080	Max. 65° F., Min. 55° F.
Refrigerator water ..	246	80	6,620	Max. 80° F.
Copper condensate flash ..	2	212	2,490	
			79,200	

We want 701 lb. of water heated to various temperatures. Let us put all this water through the fermenting vessel attemperating coils and see how things go.

Fig. 363 shows the lay-out.

If all the 701 lb. of water is put through the fermenting cooler coil it emerges at a temperature of 62° F.

The 185 lb. of gravity-adjusting liquor is then put through the cool end of the frig.

As this water enters the frig. at 62° F. we must do the final frig. cooling with additional cold water which must be run to waste as there is no use for it. We will assume that we can heat 185 lb. to 70° F. in the frig.

This adjusting liquor is then passed through a surface condenser attached to the copper, where it picks up 17,925 Btu and is raised to sterilising temperature—about 167° F.

The remaining 516 lb. of water from the fermenting cooler goes to the ammonia condenser of the refrigerator where it is heated to 74° F.

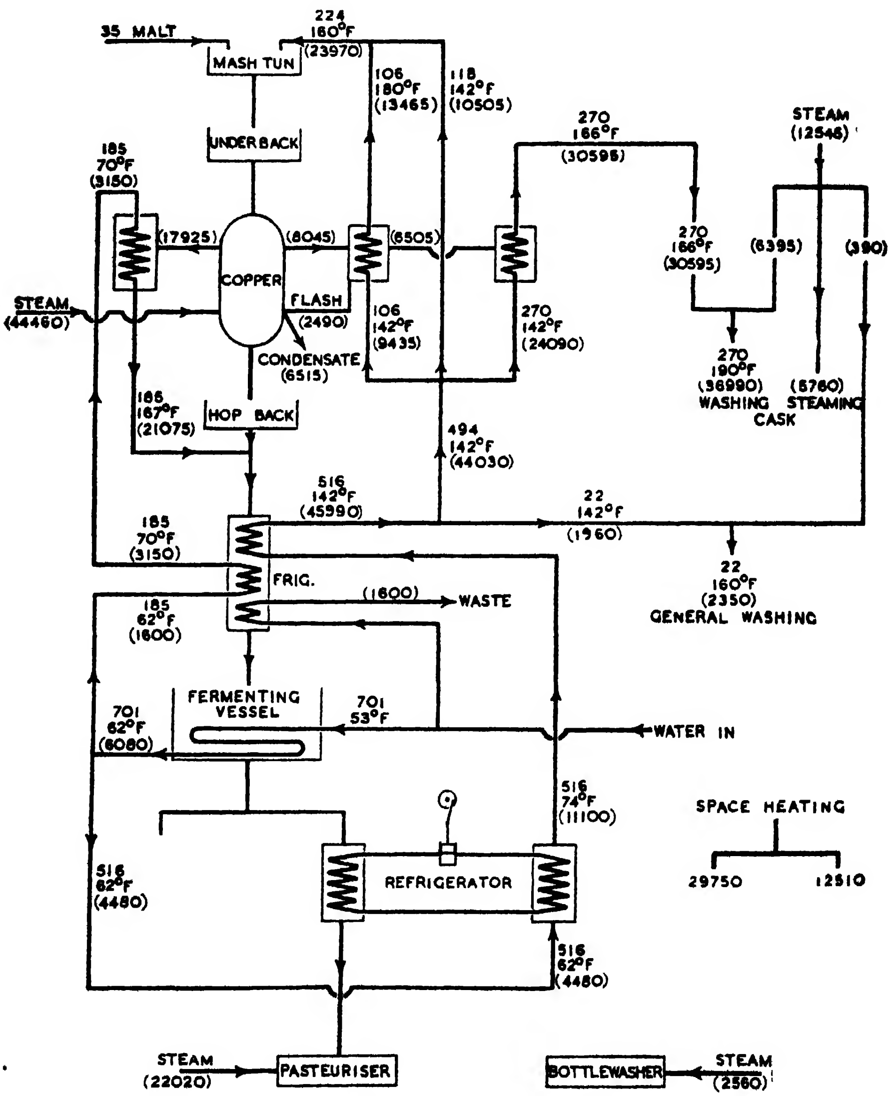


FIG. 363. BREWERY FLOW DIAGRAM WITH HEAT RECOVERY WITHOUT ANY ALTERATION TO PROCESS TECHNIQUE

It is then passed through the hot end of the frig. and is heated to 142° F. (Fig. 357 tells us that frig. water is at times heated to 150° F., so this is within practical limitations.)

Now we want 224 lb. at an average temperature of 160° F. for the mash; some of this liquor is needed at 180° F.

If 516 lb. contain 45,990 Btu leaving the frig.,
224 lb. will contain 19,970 Btu, or 89 Btu/lb.

Now we want our 224 lb. to contain 23,970 Btu. and some must be at 180° F. and contain 180 − 53 = 127 Btu/lb.

Let x be the amount of liquor at 180° F.

$$\text{So } 127x + (224 - x)\,89 = 23{,}970$$
$$38x = 4{,}035$$
$$x = 106 \text{ lb.}$$

We therefore pass 106 lb. through another surface condenser on the copper vapour and heat this up to 180° F. The remaining 118 lb. is passed direct to the mashing department at 142° F. By suitable mixing the desired mashing temperatures can be achieved.

The 270 lb. of cask washing water are passed through another surface condenser to condense the remaining copper vapour and the flash steam from the copper condensate, thus raising the wash water temperature to 166° F.

We shall need 12,545 Btu of steam heat to heat up the washing waters to the needed temperatures and to steam the casks.

Fig. 363 shows three separate surface condensers on the copper vapour. These could of course be combined into one condenser with three separate water passes.

We now have the following steam heat inputs :—

Copper	44,460	
Pasteuriser	22,020	
Washing and steaming	12,545	
Bottle washer..	2,560	
Space heating	42,260	
	123,845	
Less the returned condensate containing ..	6,515	
Total steam heat needed	117,330	Btu/barrel

The steam heat requirement without any heat recovery was 194,930 Btu/barrel. Recovery of good waste heat shows a saving of 40 per cent.

It is not suggested that this would be obtained in practice. The extra heat recovering plant would incur greater heat losses than the straight wasteful process. We might hazard a guess that if the reasonable losses are 15 per cent.

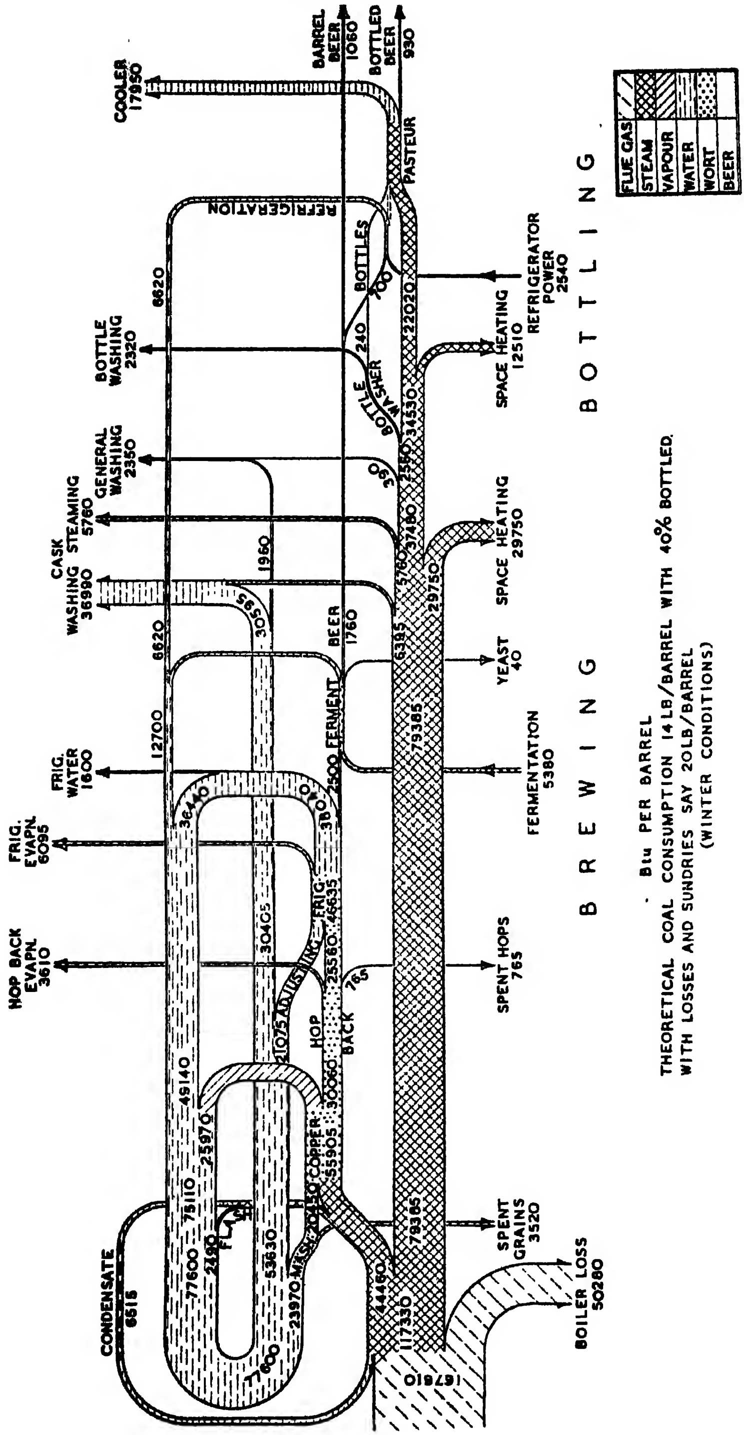


FIG. 364. BREWERY HEAT DISTRIBUTION WITH HEAT RECOVERY WITHOUT ALTERATIONS TO PROCESS TECHNIQUE

without heat recovery, they should not be more than 20 per cent. with heat recovery. The comparison then would be

Without heat recovery	..	194,930 + 15 per cent. = 224,170 Btu
With heat recovery	..	117,335 + 20 per cent. = 140,800 ,,

A saving of 38 per cent.

Suppose that we can only secure two-thirds of this, it is well worth capturing.

This heat recovery system is shown as a Sankey diagram in Fig. 364.

630. OTHER POSSIBILITIES—STERILISATION. Some brewers consider that it is not necessary to sterilise the liquor with heat, but that completely satisfactory sterilisation can be obtained cold with ozone. This would seem to be an excellent method of heat saving. So it would be if the brewery were doing no heat recovery. If, however, all or most of the heat was being recovered from the frig. nothing is gained from cold sterilisation. Some 20,000 Btu less would be put into the frig. and some 20,000 Btu less are available for recovery.

631. OTHER POSSIBILITIES — COPPER VAPOUR. Some brewers consider that actual evaporation in the copper is unnecessary, and that heating under slight pressure with adequate mechanical circulation will do all that is wanted. Here again it would seem that there is a possible saving of some 25,000 Btu. But if the vapour heat is all being recovered there is no saving. The steam that was saved from the copper will have to be used direct on the materials that were being heated by the copper vapour.

So here is an excellent example of the fundamental fact that the heat requirements of any process are primarily based on the essential sensible heating. There is no gain in saving an evaporating operation if the heat in the evaporated vapour is being recovered.

But this is not quite the whole story. Heat recovery calls for plant, some of it expensive plant. Heat recovery means not only more capital cost but more maintenance, more complicated process operation, more radiation losses, and, in a brewery, something else to keep sterile.

So we can say that it is always a good thing to cut out a process, even if the heat is being recovered, and to replace it by using direct steam elsewhere to replace the recovered heat. But this rule must be always accompanied by the postulate that the direct steam used is taking the place of direct steam saved elsewhere.

632. OTHER POSSIBILITIES—THE REJECT HEAT. If we look at Fig. 364 we see that there are two major heat rejectors. The boiler house flue gases and cask washing water. By means of a heat exchanger we could probably recover half the heat in the cask-washing water. This would heat the cold water for cask washing up to about 120° F., quite warm enough to feed into an economiser without risk of causing condensation of the flue gases. If an economiser is already fitted to the boiler we might add a sub-economiser.

The general washing water can be similarly treated.

As regards cask steaming, some brewers line their barrels with wax or enamel, thus enabling them to dispense with steaming, as good washing will completely clean the casks.

633. BOGEY. In order to ascertain bogey for this brewery—that is the reasonable target minimum heat requirements—we will assume that we can do all the following things :—

Work continuously on three shifts.

Use pressure heating in the copper without evaporation.

Prevent flash from the hopped wort by cooling the wort in the copper before relieving the pressure, with warm liquor.

Use cold-sterilisation.

Use a heat exchanger on cask washing water.

Cut out cask steaming.

Heat cask washing and general liquor in a sub-economiser.

Use covered mash tuns, hop backs, etc., all properly lagged.

This will give us the minimum heat usage for known technique and is shown in Fig. 365.

The working out will not be shown in detail ; a few points only will be dealt with.

As we are going to use heat exchange in the frig. to the greatest possible extent, we must lose no unnecessary heat by radiation from the hop back and the jack back. These and all the other plant must be properly lagged and covered.

As the copper must be well heated to ensure equivalent performance to that given by boiling, it is assumed that its contents are heated to 230° F. under a pressure of 6 psi. When the pressure is released there would be some flash evaporation. Before the pressure is released the hopped wort can be cooled by the addition of cold or tepid liquor, which will replace some of the adjusting liquor that must anyhow be added later.

It has been assumed that the low temperature end of the frig. will call for a waste of 1,000 Btu.

It may be better to replace the frig. with a multipass heat exchanger or Paraflow which can give a more effective heat exchange.

It has been assumed that we can pick up about 13,000 Btu from the cask washings heat exchanger. This is only a 35 per cent. recovery—a conservative figure.

A small heat exchanger is fitted to recover the flash from the copper heating condensate.

This leaves only 14,740 Btu to be provided by the sub-economiser, which should be easily managed. If the boiler efficiency is assumed to be 70 per cent., the saving of 14,740 Btu in the sub-economiser will increase the boiler efficiency to 81 per cent. This is by no means an unreasonable bogey with a good Lancashire boiler plant and a sub-economiser which accepts water at 145° F. and heats it only to 190° F.

Fig. 366 shows this ideal brewery as a Sankey diagram, in which about 20 per cent. extra process heat has been allowed for reasonable unavoidable losses and to provide for the small processes that have so far been ignored. This 20 per cent. must also cover the banking losses and the starting and stopping losses. This may not be enough.

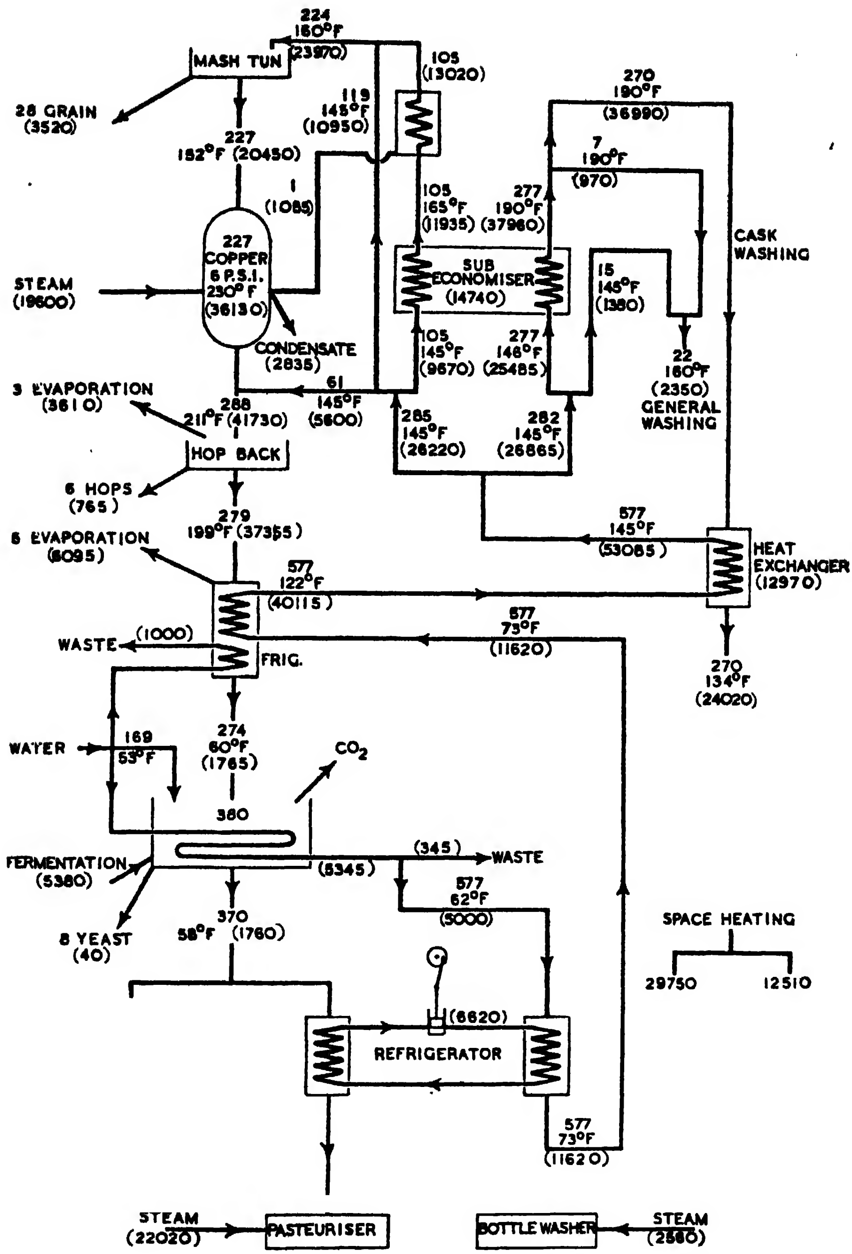


FIG. 365. BREWERY FLOW DIAGRAM WITH BOGEY HEAT REQUIREMENTS. TECHNIQUE ALTERED

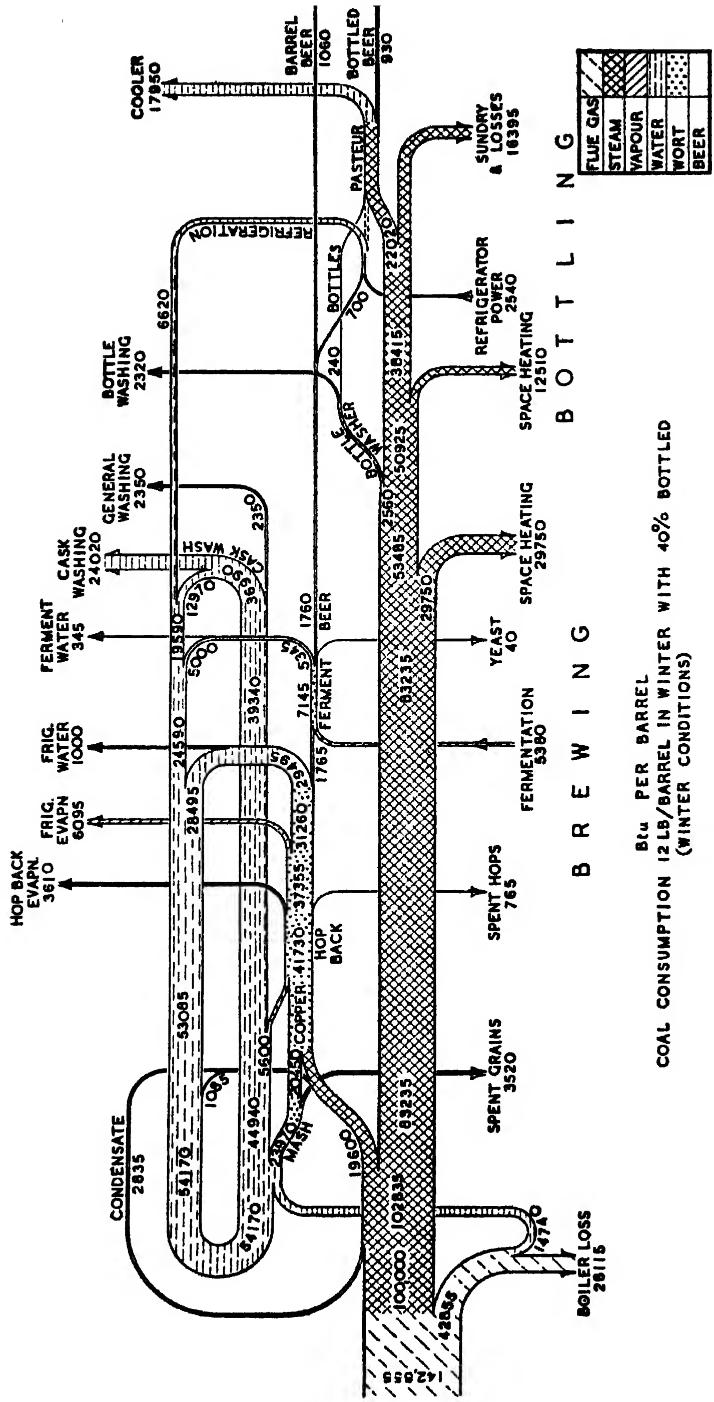


FIG. 366. BREWERY HEAT DISTRIBUTION WITH BOGEY HEAT REQUIREMENTS. TECHNIQUE ALTERED

We have now found bogey for a small brewery brewing a particular kind of beer, doing 40 per cent. of its trade in bottles and taking winter conditions.

The amount of steam heat needed is :—

	Btu/barrel
Copper (less condensate)	16,765
Space heating	42,260
Pasteuriser	22,020
Bottle washer	2,560
	83,605
Plus 20 per cent. for losses and sundries, say	16,395
	100,000

Coal—boiler efficiency, 70 per cent.	142,855 Btu/barrel.
Bogey coal consumption	12 lb./barrel.

In summer of course all the space heating heat requirements disappear ; this will make a big reduction in bogey. On the other hand there may be considerably more refrigeration to be done in summer. In a process factory it is seldom reasonable to have different bogeys for summer and winter.

All the heat recovery that has been considered is only realisable if the brewery works continuously. But why not work continuously ? For the same output the brewery would only need to be a little more than a third of the size. A few breweries work continuously, but the majority work day-work only. Brewing is one of the processes that cries out for continuous operation.

634. ACTUAL CONDITIONS. Now the brewery whose conditions were set out in Fig. 357 is doing quite a lot of heat recovery yet it is using 40 lb. coal/barrel. It condenses its copper vapour in its cask washing water. It uses its frig. cooling water for mashing. It sterilises its liquor cold. Fig. 367 shows the arrangement of its process.

The steam heat demand for all its major processes is :—

Mashing liquor	4,480
Copper	44,460
Cask washing	12,980
Cask steaming, say	5,760
General washing	390
Pasteuriser	22,020
Bottle washer	2,560
Space heating—Brewery	29,750
Bottlery	12,510
	134,910 Btu/barrel.

These are the steam heat requirements. By inspection of the boiler house we estimate that the boiler efficiency is about 63 per cent.

We can now construct Fig. 368 which is the Sankey diagram of this brewery. Like nearly all first factory heat balances it is a sorry looking picture. Out of 302,400 steam heat units we can only account for 134,910 or 45 per cent.

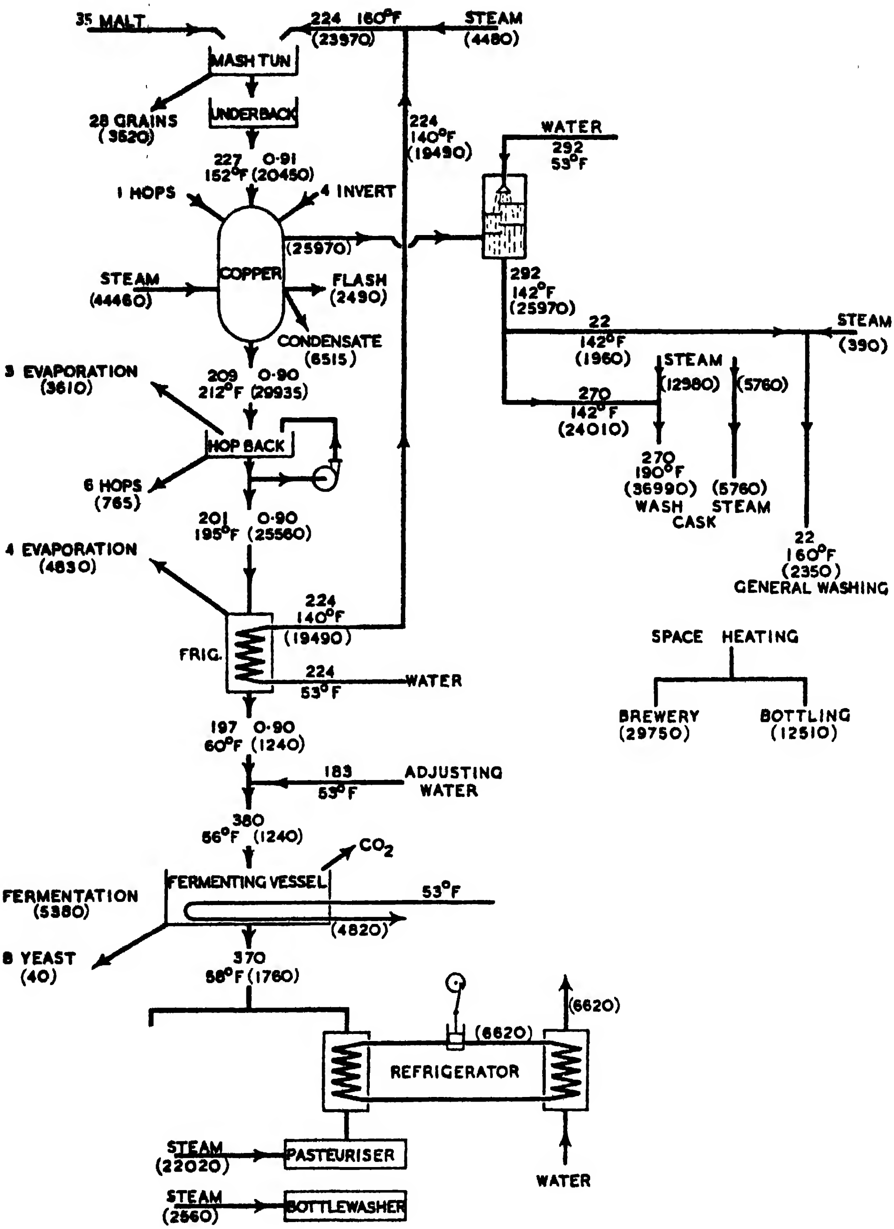


FIG. 367. FLOW DIAGRAM OF ACTUAL BREWERY

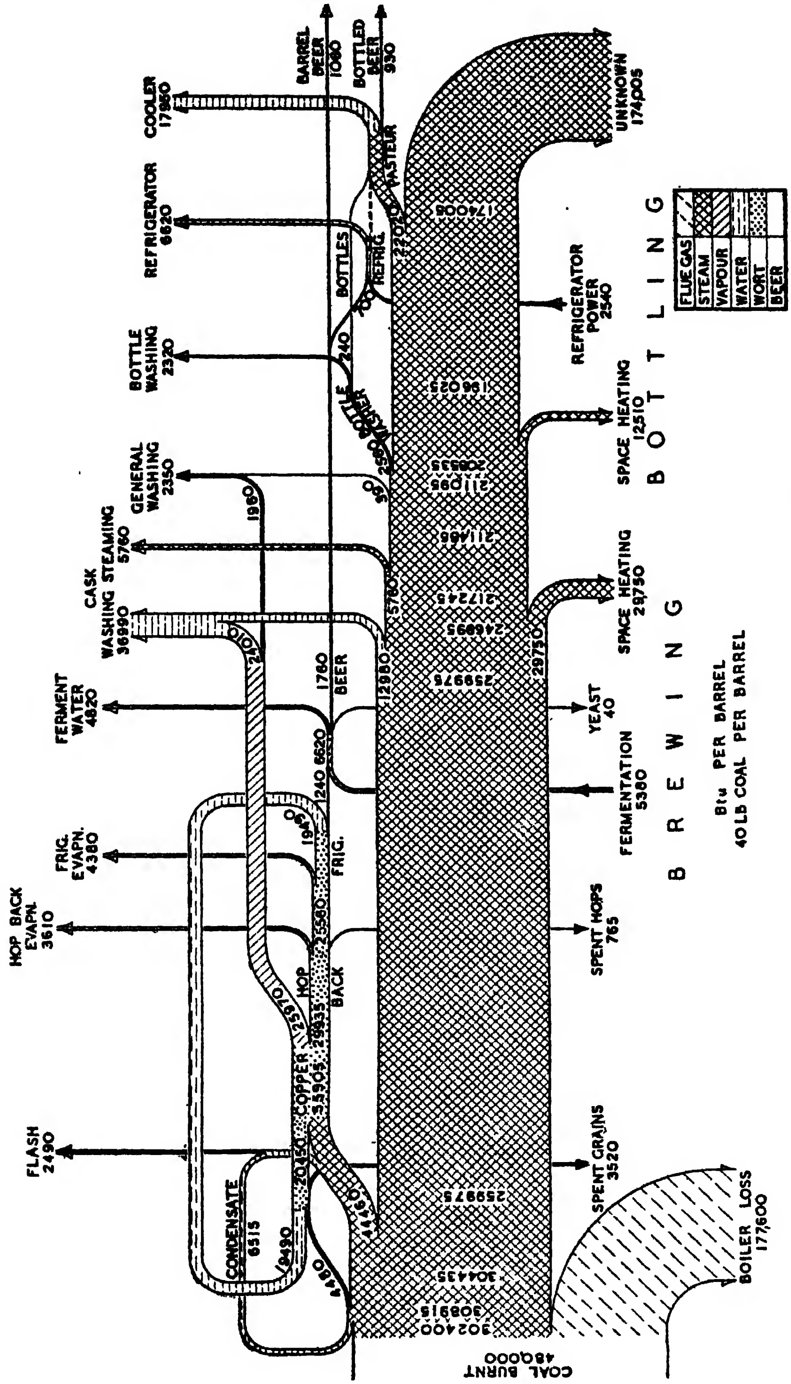


FIG. 368. HEAT DISTRIBUTION IN ACTUAL BREWERY

In this brewery 55 per cent. of the steam is going about its private business unknown to the management. It is up to the management to find out what these private affairs are, always provided it is worth while. The brewer's primary preoccupation is the quality of his beer. Provided quality control can be maintained a steam campaign is probably well worth while. It is quite a big money saver, which goes on and on and gives a good reserve against bad times.

It is probably not beyond practical possibilities to reduce the unknown steam to half the known steam. On closer investigation some of the unknown will receive promotion into the known. But because it is known that does not mean it need be used, although we know that some small processes were omitted. Do these small processes require steam ? Why is steam needed to dissolve sugar ? Because the sugar that is being used is in the wrong form.

Suppose that on a really close examination the known heat goes up from 125,905 to 150,000—this is a very generous supposition—and that the brewer will not be satisfied until his known steam consumption accounts for at least 75 per cent. of the whole. This would reduce his total steam requirements to 200,000 Btu/barrel.

He will during his heat campaign certainly screw up his boiler efficiency from 63 per cent. to 65 per cent. His coal heat requirements will be 307,700 Btu giving a coal consumption of 25·6 lb./barrel. This shows a saving of over 14 lb./barrel or 6¼ tons per week. Small breweries generally have no railway or canal facilities and their coal is expensive. The brewery whose data is set out in Fig. 357 was paying 60*s.* odd for its coal in 1944, so that a coal saving such as has just been indicated would represent £1,000 a year. These figures should be doubled for 1957 values.

As the investigation proceeds Fig. 368 will be amplified. It will probably become much more complicated. This does not matter at all. Complication in the picture can be overcome by the use of brightly coloured chalks or water colours to differentiate between steam, water, wort, etc.

All the refinements due to the different qualities can be allowed for. It may be desirable to draw one picture for the common-to-all processes, and others for each different quality. These can then all be combined finally on to one accurate diagram.

635. POWER REQUIREMENTS. Fig. 357 gives the power load as 7,000 kWh per week or 7 kWh/barrel.

During working hours, 6.0 a.m. to 5.0 p.m., 55 hours, 1,000 barrels are brewed, or 18 barrels/hr.

The load therefore is 126 kW on a brewing-hour basis.

Now in the bogey brewery 100,000 Btu of steam at 25 psi are needed.

The total heat in 25 psi steam is 1,170, and the net heat above 53° F. is 1,149 Btu/lb.

The steam used will be $\frac{100{,}000 \times 18}{1{,}149} = 1{,}567$ lb./hr.

If we raised steam at 200 psi, Table X tells us that we could expect about 480 H.P. from an ideal engine taking 10,000 lb./hr. This would be 75 H.P. from an ideal engine taking 1,567 lb./hr.

The efficiency ratio of 66 per cent. given in Table X could not be expected from such a small machine. 50 to 55 per cent. is more likely.

So that in a bogey brewery we could not expect to generate more than about 30 kW. It might not be worth while installing an electrical generating plant which could only provide quite a small part of the power needs, but it would certainly both pay and be nationally economic to drive the refrigerator by a steam engine.

The brewery is, however, nowhere near bogey. However hard it tries it will not approach bogey for 5 years or so. Let us take its present steam consumption. It is using 480,000 Btu/barrel of coal heat. At 60 per cent. boiler efficiency this will give a steam heat of 288,000 Btu/barrel. The quantity of steam will be $\dfrac{288{,}000 \times 18}{1{,}149} = 4{,}500$ lb./hr.

If this amount is raised at 200 psi, we can expect an ideal H.P. of 216 and we can hope for an efficiency ratio of 60 per cent. We could therefore produce 97 kW.

This is quite a big proportion of the load, and it would probably be well worth while installing an engine and generator to produce this load and to purchase the requirements of one department, say the bottlery, from the public supply.

There would seem however to be no very good reason for using steam at 25 to 30 psi. A little modification of heating surfaces might allow 10 psi. to work quite satisfactorily. We could then generate 127 kW—just exactly the present load.

But this postulates that the brewer is content to keep his thermal socks in folds around his ankles. If he greatly reduces his steam consumption he eliminates the possibility of being able to generate the whole of his own power.

So here is another good example of the way in which the power/steam ratio deteriorates as the thermal efficiency improves.

However, because a good brewery cannot possibly hope to generate all its own power is no reason why it should not generate some. The question of sharing load between the brewery generating plant and the outside supply is discussed in Section 804.

636. BREWERY DISCUSSION. The brewery that we have been considering uses 40 lb. coal/barrel. At 120*s.* per ton for coal this represents 2/2d per barrel. If the consumption of coal could be reduced by half, the saving would amount to £2,600 a year. One large London brewery has cut its coal consumption by more than half over ten years, so the target is not unreasonable, and the plant would not be very costly.

The brewer's chief preoccupations are the flavour, colour, brightness and keeping properties of his beer. He is very chary of making any change however economical if any of these qualities might be affected. The technique suggested in Fig. 366 does postulate a change in process to which he might take exception. On the other hand there is no process change in the arrangement shown in

Fig. 364, in fact the copper vapour has been condensed in a surface heater so that no hop oils or other volatiles can be absorbed by the heated water.

All the Sankey diagrams of the brewery show how extravagant in heat the bottling process is. Take the figures from Fig. 366 :—

The heat requirements for bulk beer are

	Btu/Barrel
Process steam (less condensate) ..	16,765
Sub-economiser	14,740
Space heating	29,750
	61,255

The heat requirements for bottling only are

	Btu/Barrel	
Washing	2,560	
Pasteurising	22,020	
Space heating	12,510	
	37,090	for 40 per cent.
This becomes	92,725	for 100 per cent.

Therefore, bottling adds 150 per cent. to the bulk beer heat requirements, of which pasteurising is responsible for three-fifths.

We see in the brewery, as in the laundry, that space heating is the biggest single heat user. In the design of the average factory building the question of loss of heat from the building is seldom considered. The single storey light roof modern building is a great offender. By lining the roof with fibreboard a very large and lasting economy can be cheaply made and the resulting building will be much more comfortable to work in. Similarly, the venting of steam, or gas or smells is too often done by greatly increasing the general ventilation causing a huge air current through the factory and all this air must be heated. If the steam, gas or what-not is removed locally it is often possible to secure another great and lasting saving.

Objection may be taken to the brewery that has been discussed because it is somewhat unorthodox, and because some of the suggestions in the foregoing sections are still more unorthodox. A bogey has therefore been worked out for a brewery, using orthodox methods, doing only one brew per day, having no cool well water available, doing hot liquor sterilisation, brewing only strong beer, evaporating 12 per cent. in the copper, allowing over-generous heat losses, but doing no bottling. The process is shown in Fig. 368a and its details are as follows :—

All mashing liquor raised to 212° F. No breakdown or dilution. Liquor from Mains at 65° F. Unlined casks.

Space Heating	Requirements	30,000 Btu/Barrel
	Available heat losses	33,730 „ „
Process ..	1 Brewing of 200 barrels/day.	

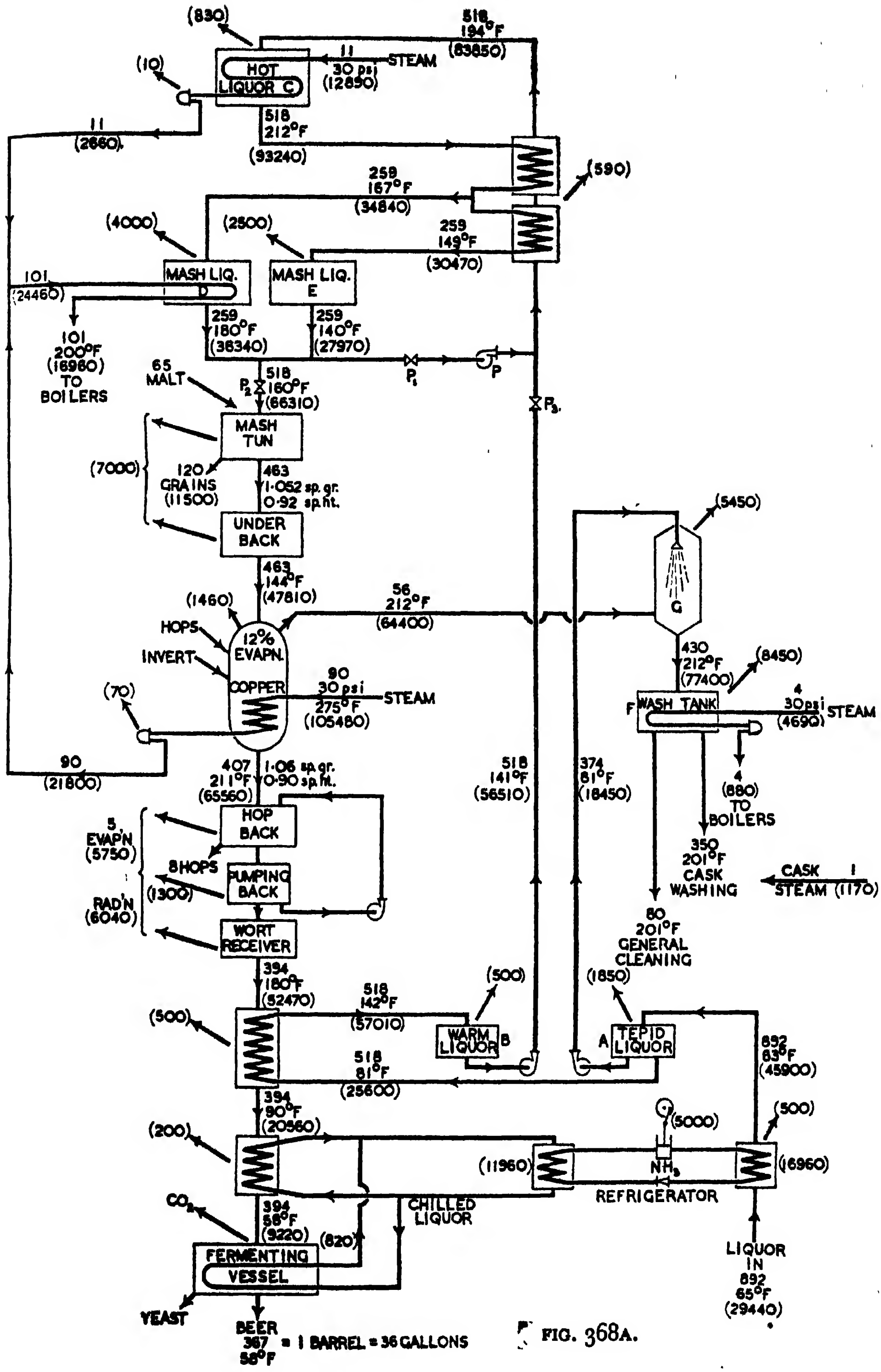

(830)
518
194°F
(83850)
(10)
HOT
LIQUOR C
11
30 psi
(12890)
STEAM
11
(2660)
518
212°F
(93240)
(590)
259
167°F
(34840)
(4000)
(2500)
259
149°F
(30470)
MASH LIQ.
D
MASH LIQ.
E
101
(24460)
101
200°F
(16980)
TO
BOILERS
259
180°F
(36340)
259
140°F
(27970)
P_1
P
65
MALT
P_2
518
160°F
(66310)
P_3
MASH
TUN
(7000)
120
GRAINS
(11500)
463
1·052 sp. gr.
0·92 sp. ht.
UNDER
BACK
(5450)
463
144°F
(47810)
56
212°F
(64400)
(1460)
HOPS
INVERT
12%
EVAPN.
COPPER
G
430
212°F
(77400)
(8450)
(70)
90
30 psi
275°F
(105480)
STEAM
F
WASH TANK
4
30psi
(4690)
STEAM
90
(21800)
407
211°F
(65560)
1·06 sp. gr.
0·90 sp. ht.
518
141°F
(56510)
374
81°F
(18450)
4
(880)
TO
BOILERS
5
EVAP'N
(5750)
HOP
BACK
8 HOPS
(1300)
PUMPING
BACK
350
201°F
CASK
WASHING
CASK
STEAM
1
(1170)
RAD'N
(6040)
WORT
RECEIVER
80
201°F
GENERAL
CLEANING
394
180°F
(52470)
518
142°F
(57010)
(500)
(1850)
(500)
WARM
LIQUOR
B
A
TEPID
LIQUOR
892
83°F
(45900)
518
81°F
(25600)
394
90°F
(20560)
(5000)
(500)
(200)
(11960)
NH_3
(6960)
REFRIGERATOR
CO_2
394
58°F
(9220)
(820)
CHILLED
LIQUOR
FERMENTING
VESSEL
LIQUOR
IN
892
65°F
(29440)
YEAST
BEER
367
58°F
= 1 BARREL = 36 GALLONS

FIG. 368A.

The whole liquor input goes through the refrigerator condenser and flows only when the refrigerator is running.

The refrigerator runs intermittently during most of the 24 hours to maintain the chill on the attemperating liquor which cools the fermenting vessel. (The chilled liquor tank is not shown.) When wort is being cooled in the paraflow or heat exchanger the refrigerator runs continuously.

The liquor enters the brewery at 65° F. and picks up 18° F. from the refrigerator, which thus runs as a heat pump.

From the refrigerator condenser the liquor runs to the tepid liquor tank A which has a capacity of 430 barrels (20 hours).

From the tepid liquor tank A the washing water is pumped through the spray condenser G while the copper is boiling into the wash tank F. Tank F must have a capacity of about 23 hours or 250 barrels. At its high temperature of over 200° F. its heat loss will call for a little steam heat.

From the tepid liquor tank A the mashing liquor runs through the wort paraflow into the warm liquor tank B which can be quite small as it is only a pump supply tank.

From tank B the mashing liquor is pumped through the liquor paraflow or heat exchanger into the hot liquor tank C. Tank C is quite small, being simply large enough to accommodate the necessary heating surface. In tank C the liquor is brought to 212° F. and runs back through the liquor paraflow, giving up much of its heat to the incoming warm liquor.

From the paraflow the liquor goes to the mashing liquor tanks D and E where it remains until required for mashing.

The liquor in tank D is brought to 180° F. and its heat loss made good by the condensate coil which cools the condensate from the copper and tank C, thereby avoiding the need for any flash collection system.

Tanks D and E must be full mashing capacity, or 150 barrels each.

After the week-end stop the liquor in tanks D and E will be too cool for Monday's mash. So valves P2 and P3 are closed, valve P1 is opened and pump P is started, the liquor is then circulated through tank C until it has been sufficiently heated.

Heat Losses .. For lagged surfaces 0·5 Btu/sq. ft./° F./hr.
For bare surfaces 2·0 Btu/sq. ft./° F./hr.

In order to be very conservative and to allow for too great a heat loss, the initial temperature differences will be taken, not the diminishing temperature difference as discussed in Section 464.

Tank A .. 20 hours capacity is 2,400 cu. ft.
2 tanks each 8 ft. × 8 ft. × 20 ft. give 2,560 cu. ft.
Lagged surface = 768 sq. ft. each.
Temperature difference : 20° F. for 24 hours.
Total loss : 368,000 Btu/Day or 1,843 Btu/Barrel.

Tank C .. 1 hour capacity is 70 cu. ft.
1 tank 4 ft. × 4 ft. × 6 ft. = 96 cu. ft.
Lagged surface : 128 sq. ft.
Temperature difference : 150° F. for 6 hours.
Total loss : 57,600 Btu/Day or 288 Btu/Barrel.
Add about 200 per cent. for rapid circulation : 830 Btu/Barrel.

Tanks D and *E.* Mashing capacity Half each or 832 cu. ft.
Each tank 6 ft. × 8 ft. × 18 ft. = 860 cu. ft.
Lagged surface : 600 sq. ft. each.
Temperature difference :
Tank "D" 120° F.
Tank "E" 80° F. } for 20 hours.
Total loss :
Tank "D" 720,000 Btu/Day or 3,600 Btu/Barrel.
Tank "E" 480,000 Btu/Day or 2,400 Btu/Barrel.

Tank F .. 24 hour capacity is 1,380 cu. ft.
2 tanks each 8 ft. × 8 ft. × 12 ft. give 1,540 cu. ft.
Lagged surface : 512 sq. ft. each.
Temperature difference : 145° F. for 22 hours.
Total loss : 1,633,280 Btu/Day or 8,166 Btu/Barrel.

Copper .. Capacity : 450 Barrels or 2,600 cu. ft.
12 ft. diameter or 20 ft. high over cylindrical part.
Surface of dome : 230 sq. ft.
Surface of rest : 900 sq. ft.
Heat loss of bare dome : 2 Btu/sq. ft./° F./hr.
Temperature difference : 150° F. for 3½ hours.
Total loss :
Bare dome 241,500 Btu/Day or 1,210 Btu/Barrel.
Rest 236,250 Btu/Day or 1,180 Btu/Barrel.
Total : 2,390 Btu/Barrel.

By lagging the dome the copper steam consumption would be reduced by 1 lb./Barrel, and the copper heat loss would be reduced to about 1,460 Btu/Barrel.

Remaining vessels have had losses apportioned on basis of the foregoing allowing a good margin for pipe losses.

Power Generation.—The steam requirements are so small and irregular that it would be out of the question to try to generate any power.

Steam requirements .. .. 106 lb./Barrel

Coal needed at 70 per cent. boiler efficiency.. .. .. 13 lb./Barrel

Add 33 per cent. for banking and week-end .. 17½ lb./Barrel

637. FOOD EXTRACT. The object of including this example is fourfold. First, because it shows that a plant that seems very efficient according to certain lights may prove to be very inefficient if the lights are changed. Second, because the main part of the process is evaporation—as distinct from drying—and evaporation is more of a science than an art. Third, because it shows how technique from one industry can often be adapted to another if only the knowledge of the goings-on in other industries were more generally sought. Fourth, because it shows what a tremendous amount of information can be extracted from apparently meagre data.

The visit to the factory was brief and figures for the main processes only were taken. The omission of details for the other processes makes no difference to the main argument as will be seen.

The product is a syrupy extract containing 17 per cent. of water and amounts to 290 tons a week.

The raw material is 440 tons of meal a week containing 15 per cent. of moisture.

The exhausted meal leaves the process with 60 per cent. of water.

It is dried by steam-heated hot air down to 11 per cent. moisture and then weighs 133 tons per week, and is sold for cattle food.

The factory has five Lancashire boilers, two of which are fitted with economisers.

Inspection of the boiler plant suggests that, though the plant is well cared for, its efficiency is unlikely to reach 65 per cent.

The steam pressure is 120 psi.g.

The steam is passed through engines, reputed to use 35 lb./kWh, and exhausts to process at 8 psi.g.

The electrical load is 500 kW.

The coal, of good quality, burnt per week is 210 tons.

The condensate return system is excellent, but all the flash is lost.

The hours worked are 96 per week.

Both double effect evaporators work under the same vacua, namely 15 in. and 27 in.

638. PROCESS. The meal is steeped in water at about 150° F.

The thin extract is run off, chemically treated, clarified and filtered. It then contains an average of 72 per cent. of water.

The thin extract is preconcentrated in a double effect evaporator to 52 per cent. of water.

Process washings at about 170° F. and containing about 93 per cent. of water are thickened up to 72 per cent. water in a double effect evaporator, and are then mixed with the thin extract.

The amount of washings is unknown, but a rough measurement of the condensate from the first effect of the washings evaporator gave 3,500 lb. condensate/hour.

The preconcentrated extract is finally concentrated to 17 per cent. water in a single effect evaporator.

Rough particulars of the size and construction of the main buildings were taken.

639. ELABORATION OF DATA. Now that is all the data we possess. Let us elaborate it to the utmost and see how much information we really have got. We will base all figures on 1 lb. of extract solids as this will avoid all kinds of complications.

PER WEEK

	Sample tons	*Per cent. water*	*Water tons*	*Solids tons*
Raw material ..	440	15	66	374
Extract	290	17	49	241
Dried meal ..	133	11	15	118

640. CONCENTRATION OF WASHINGS. We can solve the double effect evaporator problem with only one known quantity with the aid of Fig. 369. We know the vacua in the two effects and we know the input steam pressure. Boiling point elevation, which will be very small in this operation, will be ignored.

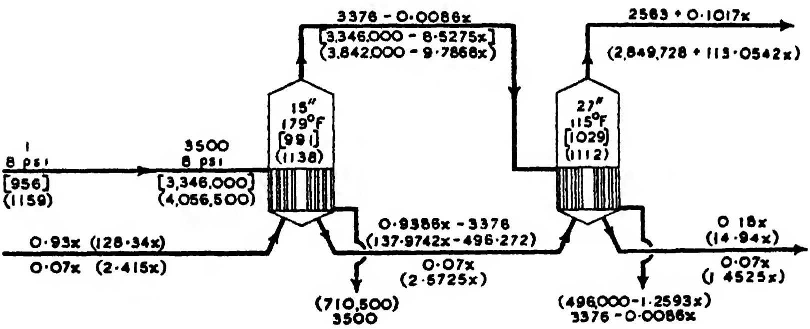


FIG. 369. DOUBLE EFFECT WASHINGS EVAPORATOR—ANALYSIS

The condensate from the first body was 3,500 lb./hr. Table IV tells us that 8 psi condensate will lose 2·37 per cent. when its pressure is reduced to atmospheric. As the steam was almost certainly more than 2·37 per cent. wet, we can ignore quite safely the loss by flash.

The steam input must therefore be 3,500 lb./hr at 8 psi.g. containing

Total heat 3,500 × 1,159 = 4,056,500 Btu
Latent heat 3,500 × 956 = 3,346,000 „
Sensible heat 3,500 × 203 = 710,500 „

Only the latent heat is used. The condensate carries away all the sensible heat.

The feed is x and contains $\cdot 93x$ of water and $\cdot 07x$ of solids. The specific heat of the solids is taken at ·25.

(87664) X* 2

The feed is at 170° F. and therefore contains

in the water	$\cdot 93x \times 138 =$	$128\cdot 34x$ Btu
in the solids	$\cdot 07x \times \cdot 25 \times 138 =$	$2\cdot 415x$

(As we do not yet know the magnitude of things we will compromise by taking four or five significant figures.)

Before evaporation can begin, the feed must be heated to the boiling temperature 179° F.

This calls for

$$\left.\begin{array}{r}\cdot 93x \times 9\\ \cdot 25 \times \cdot 07x \times 9\end{array}\right\} 8\cdot 5275x \text{ Btu}$$

The heat available for evaporation is therefore $3{,}346{,}000 - 8\cdot 5275x$.

Each pound of 15-in. vapour calls for 991 Btu of latent heat.

The evaporation will be

$$\frac{3{,}346{,}000 - 8\cdot 5275x}{991} = 3{,}376 - \cdot 0086x = \text{A}$$

This vapour will contain

Latent heat	=	$3{,}346{,}000 - 8\cdot 5275x$
A × 1,138 Total heat	=	$3{,}842{,}000 - 9\cdot 7868x$
Sensible heat by difference	=	$496{,}000 - 1\cdot 2593x$

The sensible heat goes out of the second effect heating surface as condensate.

The feed to the second body consists of

	$\cdot 93x - 3376 + \cdot 0086x =$	$\cdot 9386x - 3{,}376$	lb. of water
and		$\cdot 07x$	lb. of solids
containing	$(\cdot 9386x - 3{,}376)\ 147 =$	$137\cdot 9742x - 496{,}272$ Btu	
+	$\cdot 07x \times \cdot 25 \times 147 =$	$2\cdot 5725x$ Btu	

Check across the first effect

	Weight			*Heat*		
In	3,500		$\cdot 93x$	4,056,500		$128\cdot 340x$
			$\cdot 07x$			$2\cdot 415x$
	3,500	+	x	4,056,500	+	$130\cdot 755x$
Out	+ 3,376	−	$\cdot 0086x$	+ 3,842,000	−	$9\cdot 7868x$
	− 3,376	+	$\cdot 9386x$	− 496,272	+	$137\cdot 9742x$
		+	$\cdot 0700x$		+	$2\cdot 5725x$
	+ 3,500			+ 710,500		
	3,500	+	x	4,056,228	+	$130\cdot 7599x$

The small discrepancies are clearly of no importance.

The output from the second body is to be 72 per cent. water and 28 per cent. solids.

The input to the first body contained 93 per cent. water and 7 per cent. solids.

The output from the second body will be 18 parts water and 7 parts solids.

The output weight will be $\cdot 18x$ water and $\cdot 07x$ solids.

At 27 in. vacuum the sensible heat is 83 Btu/lb. so that the concentrated washings will contain

in the water $\cdot 18x \times 83 = 14\cdot 94x$ Btu/lb.
in the solids $\cdot 25 \times \cdot 07x \times 83 = 1\cdot 4525x$ Btu/lb.

The heat input to the second effect is	*Btu*
Vapour	3,842,000 − 9·7868x
Feed	− 496,272 +137·9742x
	3,345,728 +128·1874x

The known heat output is	
Condensate	496,000 − 1·2593x
Thick washings water	+ 14·9400x
Thick washings solids	+ 1·4525x
	496,000 + 15·1332x

Therefore the heat in the second effect vapour is
2,849,728 +113·0542x Btu

This figure divided by 1,112 (the total heat in 27-in. vapour) gives the weight of the second effect evaporation, or 2,563 + ·1017x lb.

We can now take a balance over the second effect and can equate for x

	Weight	*Heat*
In	+ 3,376 − ·0086x	+3,842,000 − 9·7868x
	− 3,376 + ·9386x	− 496,272 +137·9742x
	+ ·0700x	+ 2·5725x
	x	3,345,728 +130·7599x
Out	+ 2,563 + ·1017x	+2,849,728 +113·0542x
	+ 3,376 − ·0086x	+ 496,000 − 1·2642x
	+ ·1800x	+ 14·9503x
	+ ·0700x	+ 1·4525x
	5,939 + ·3431x	3,345,728 +128·1928x

$$x = 5{,}939 + \cdot 3431x$$

$$\cdot 6569\,x = 5{,}939$$

$$x = 9{,}041 \text{ lb.}$$

The discrepancy on the heat balance can now be valued and amounts to

$$(130 \cdot 7599 - 128 \cdot 1928)\ 9{,}041 = 23{,}209$$

on a total of roughly

$$3{,}346{,}000 + (130 \times 9{,}000) = 4{,}516{,}000$$

Just over ½ per cent. The value of x is therefore quite near enough.

The washings amount to 9,041 lb./hr.

The weekly output of extract solids is 241 tons in 96 hours

or
$$\frac{241 \times 2{,}240}{96} = 5{,}623 \text{ lb./hr.}$$

In order to get the washings evaporator in terms of one pound of extract solids all we have to do is to substitute 9,041 for x in Fig. 369 and divide by 5,623. This has been done and the result is given in Fig. 370.

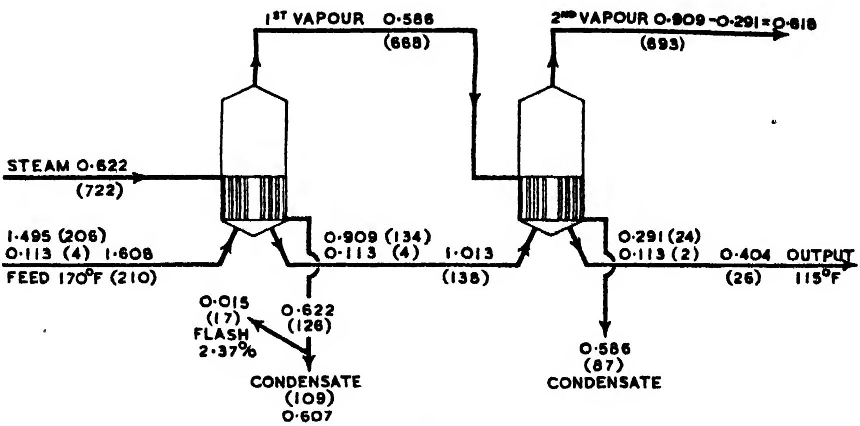


FIG. 370. DOUBLE EFFECT WASHINGS EVAPORATOR—SOLUTION

It will always be necessary to adjust the figures by 1 or 2 in the last figure to get the diagram to tie up. This is because the steam tables are only approximate and because our first analysis was an approximation only. Such cookery is of no significance. Thus the first substitution for x gives a heat input into the first effect of 932 and an output of 930. We therefore add 2 to the output in such a way as not to improve matters, namely by adding it to the flash that is lost. Similarly in the second effect we put the tie-up adjustments on to the reject vapour.

641. PRECONCENTRATION OF EXTRACT. This is done in a double effect evaporator working under the same pressure conditions as that used for the washings. The input consists of thin extract and thickened washings. Their composition is :—

72 per cent. water and 28 per cent. solids

or 2·571 lb. water and 1 lb. solids.

Total 3·571 lb./lb.E.S. (E.S. stands for extract solids.)

Of this ·404 lb. is thickened washings, so that 3·167 lb. is thin extract, of which 2·280 lb. is water and ·887 lb. is solids.

The thin extract and the thickened washings can be assumed to enter the evaporator at 110° F.

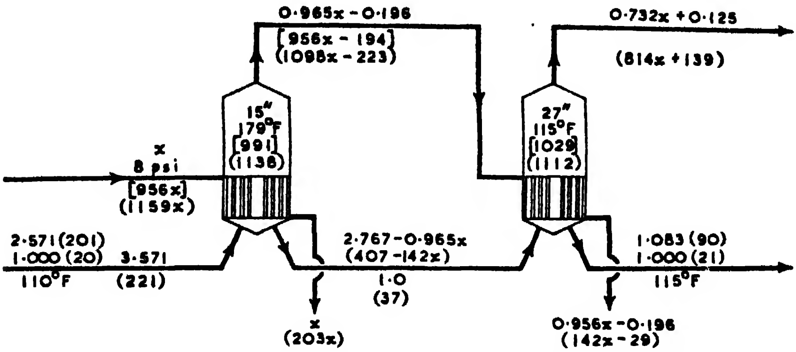


FIG. 371. DOUBLE EFFECT EXTRACT PRECONCENTRATOR—ANALYSIS

We can now construct Fig. 371.

In Fig. 369 we knew the quantity of input steam and had to find the amount of input washings. In Fig. 371 we know the amount of the feed and must find the amount of steam required. We will work the analysis out in detail in the remainder of this Section.

The input feed will contain

$$2{\cdot}571 \times (110 - 32) = 201 \text{ Btu}$$
$$1 \times {\cdot}25 \times (110 - 32) = 20 \text{ ,,}$$
$$\overline{221 \text{ ,,}}$$

The output at 115° F. consists of 52 per cent. water and 48 per cent. solids or 1·083 lb. water and 1 lb. solids (Sections 637 and 638)

containing $1{\cdot}083 \times (115 - 32) = 90 \text{ Btu}$

and $1 \times {\cdot}25 \times (115 - 32) = 21 \text{ ,,}$

$$\overline{111 \text{ ,,}}$$

Before any evaporation can start the feed must be heated from 110° F. to 179° F., calling for :—

$$2{\cdot}571 \times (179 - 110) = 177 \text{ Btu}$$
$$1 \times {\cdot}25 \times (179 - 110) = 17 \text{ ,,}$$
$$\overline{194 \text{ Btu}}$$

If x is the weight of steam fed into the first calandria

The heat available for evaporation will be $956x - 194$ Btu.

This latent heat will be the latent heat of the vapour whose total heat will be

$$\frac{(956x - 194)\ 1{,}138}{991} = 1{,}098x - 223 \text{ Btu}$$

and whose weight will be

$$\frac{1{,}098x - 223}{1{,}138} = \cdot 965x - \cdot 196 \text{ lb.}$$

The feed entering the second effect will be

$$2\cdot 571 - \cdot 965x + \cdot 196 = 2\cdot 767 - \cdot 965x$$

containing $(2\cdot 767 - \cdot 965x)\ 147 = 407 - 142x$ Btu in the water

and $1 \times \cdot 25 \times 147 = 37$ Btu in the solids

Check across first effect

	Weight—lb.	*Heat—Btu*
In	x	$1{,}159x$
	$3\cdot 571$	221
	$3\cdot 571 + x$	$221 + 1{,}159x$
Out	x	$203x$
	$2\cdot 767 - \cdot 965x$	$407 - 142x$
	$1\cdot 0$	37
	$- \cdot 196 + \cdot 965x$	$- 223 + 1{,}098x$
	$3\cdot 571 + x$	$221 + 1{,}159x$

The heat input to the second effect is

$$1{,}098x - 223 + 407 - 142x + 37 = 956x + 221 \text{ Btu}$$

The heat in the condensate and in the thickened extract must be

$$142x - 29 + 111 = 142x + 82 \text{ Btu}$$

So that the heat in the second effect vapour must be

$$956x + 221 - 142x - 82 = 814x + 139 \text{ Btu}$$

Therefore the weight of reject vapour must be

$$\frac{814x + 139}{1{,}112} = \cdot 732x - \cdot 125 \text{ lb.}$$

We can now balance over the second effect and equate for x.

	Weight—lb.	*Heat—Btu*
In	$+ \cdot 965x - \cdot 196$	$+ 1{,}098x - 223$
	$- \cdot 965x + 2\cdot 767$	$- 142x + 407$
	$+ 1\cdot 0$	$+ 37$
	$3\cdot 571$	$956x + 221$

Out	$+ \cdot 732x + \cdot 125$	$+ \ 814x + 139$
	$+ 1 \cdot 083$	$+ \ 90$
	$+ 1 \cdot 0$	$+ \ 21$
	$+ \cdot 956x - \cdot 196$	$+ \ 142x - 29$
	$1 \cdot 688x + 2 \cdot 012$	$956x + 221$

$$1 \cdot 688x + 2 \cdot 012 = 3 \cdot 571$$
$$x = \cdot 924 \text{ lb.}$$

We can now substitute ·924 for x and get Fig. 372.

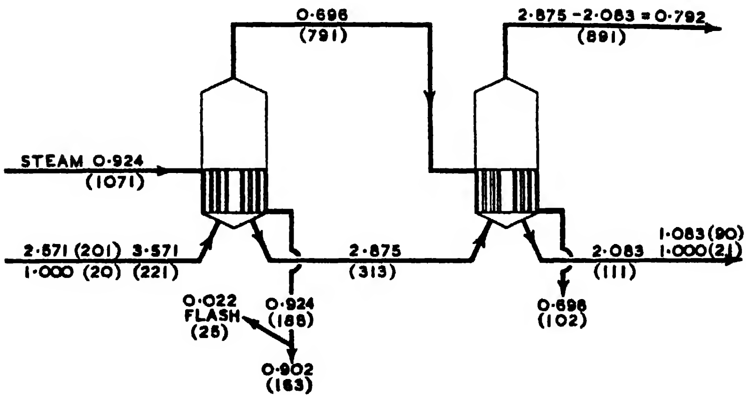


FIG. 372. DOUBLE EFFECT EXTRACT PRECONCENTRATOR—SOLUTION

The input quantities are

Steam	·924 lb.	with	1,071 Btu
Feed	3·571 „	with	221 „
	4·495 lb.		1,292 Btu

The output quantities are

First condensate ..	·902 lb.	163 Btu
Flash	·022 „	25 „
Second condensate ..	·696 „	102 „
Thickened extract ..	2·083 „	111 „
Vapour	·792 „	891 „
	4·495 lb.	1,292 Btu

642. CONCENTRATION OF EXTRACT. The final concentration is done in single effect at 27 in. vacuum.

The input thickened liquor has 52 per cent. water and 48 per cent. solids.

The concentrated extract must be 17 per cent. water and 83 per cent. solids.

The feed consists of 1·083 lb. water and 1·0 lb. solids.

The output consists of ·205 lb. water and 1·0 lb. solids.

The evaporation must be 1·083 − ·205 = ·878 lb. water.

Say the feed enters at 110° F., then it will contain

$$1 \cdot 083 \times 78 = 84 \text{ Btu in the water}$$
$$1 \times \cdot 25 \times 78 = 20 \text{ Btu in the solids}$$

The boiling point of water at 27 in. vacuum is 115° F. At the high concentration of the output extract it will be necessary to allow for boiling point elevation. As the solids in the extract are chiefly carbohydrates we can take the B.P.E. of sugar solutions as preliminary guides. At 27 in. and 83 per cent. solids the B.P.E. of a sugar solution would be 16° F.

Therefore the solids must be heated to 115 + 16 = 131° F. and the water in the output will also be at this temperature. So we can start building up Fig. 373.

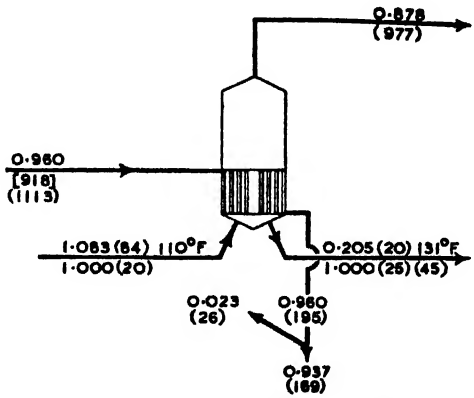


FIG. 373. SINGLE EFFECT FINAL EXTRACT EVAPORATOR

The output thick extract will contain

$$\cdot 205 \times 99 = 20 \text{ Btu in the water}$$
$$1 \cdot 0 \times \cdot 25 \times 99 = 25 \text{ Btu in the solids}$$

The evaporation to be done is 1·083 − ·205 = ·878 lb.

·878 lb. of vapour at 27 in. vacuum contains 977 Btu.

The difference between the input feed and output extract heats is 84 + 20 − 45 = 59 Btu.

The latent heat of the input steam must be 977 − 59 = 918 Btu.

The weight of the input steam must be $\frac{918}{956}$ = ·960 lb.

The total heat of the input steam will be 1,159 × ·96 = 1,113 Btu.

643. DRYING EXHAUSTED MEAL. From 374 tons of meal solids

241 are extracted and go into the finished product
118 remain in the reject meal
15 must therefore be lost.

This shows a processing loss of 4 per cent. on input or 6·2 per cent. on output. (It might be more important to try to save this loss than to cut the steam consumption).

The solids in the exhausted meal are $\frac{118}{241} = \cdot 490$ lb./lb. E.S.

The exhausted meal contains 60 per cent. water and 40 per cent. solids or ·735 lb. water and ·490 lb. solids.

The dried meal contains 11 per cent. water and 89 per cent. solids or ·061 lb. water and ·490 lb. solids.

The drying evaporation is $\cdot 735 - \cdot 061 = \cdot 674$ lb./lb. E.S.

The meal is dried in a steam heated drier. The efficiency of these machines lies between 20 and 60 per cent. As this was not a very good plant we will guess the efficiency at 35 per cent.

Only the latent heat of the drier input steam will be used.

We can assume that no heat need be added to the meal solids.

The water in the meal enters the drier at say 100° F.

It leaves as vapour with, say, 1,150 Btu/lb. (It is very difficult to estimate what the total heat/lb. of vapour will be. It depends on the partial pressure of the steam in the steam-air mixture.)

The heat output will be	$\cdot 674 \times 1150 =$	775	Btu in the vapour.
	$\cdot 49 \times \cdot 25 \times (100-32) =$	8	,, ,, ,, meal solids.
and	$\cdot 061 \times (100-32) =$	4	,, ,, ,, moisture.
Total		787	,,

If the drier is 35 per cent. efficient, the heat input must be

$$\frac{787}{\cdot 35} = 2{,}248 \text{ Btu.}$$

The wet meal brings in

$$\cdot 735 \times (100-32) + \cdot 49 \times \cdot 25 \times (100-32) = 58 \text{ Btu.}$$

So that the latent heat that must be provided by the input steam will be

$$2{,}248 - 58 = 2{,}190 \text{ Btu.}$$

or 2,655 Btu of total heat

and the steam will weigh 2·290 lb.

We can put this information down in Fig. 374.

644. HEATING RAW MATERIALS. Although we have assumed that the exhausted meal leaves at 100° F., it was at one time heated to 150° F. and would have been rejected at that temperature had it not been for losses. We will assume that it must be heated from 50° F. to 150° F.

Raw material to produce 241 tons of extract solids is 440 tons with 66 tons of water and 374 tons solids.

Raw material to produce 1 lb. of extract solids is 1·826 lb. with 0·274 lb. of water and 1·552 lb. solids.

The heat in the cold raw material at 50° F. is

$$(1{\cdot}552 \times 18 \times {\cdot}25) + ({\cdot}274 \times 18) = 12 \text{ Btu.}$$

The heat in the hot meal at 150° F. is

$$(1{\cdot}552 \times 118 \times {\cdot}25) + ({\cdot}274 \times 118) = 78 \text{ Btu.}$$

Heat to be added .. 66 Btu/lb. E.S.

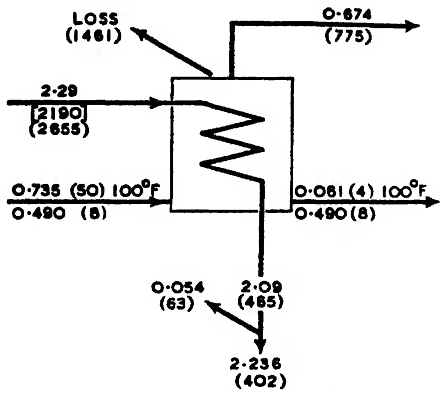


FIG. 374. EXHAUSTED MEAL DRIER

645. HEATING PROCESS WATER. The process water to be heated must consist of all the water that is put out either as liquid or as vapour. We can tabulate thus :—

Dried meal moisture (Fig. 374)	·061 lb.
Meal drier evaporation (Fig. 374)	·674 „
Finished extract water (Fig. 373)	·205 „
Washings evaporation (Fig. 370)	1·204 „
Preconcentrator evaporation (Fig. 372) ..	1·488 „
Final evaporation (Fig. 373)	·878 „
	4·510 lb./lb. E.S.

Of this 1·204 lb. must be heated to 170° F. needing	166 Btu.
and the rest, 3·306 lb. must be heated to 150° F. needing	390 „
	556 Btu/lb. E.S.

The raw meal entered at 50° F. with	·274 lb.	containing	5 Btu.
The second effect condensate is (Section 647) ..	1·153 „	„	151 „
	1·427 „	„	156 Btu
The raw cold water must be 4·510 — 1·427 =	3·083 lb.	„	55 „
			211 Btu/lb. E.S.
The heat needed for water heating is 556 — 211 =			345 Btu/lb. E.S.

646. **SPACE HEATING.** There are three main buildings that require heating. They are steel framed, brick panelled buildings with boarded and slated roofs. The windows were estimated at 20 per cent. of the walls. The smaller packing building is the only one provided with roof lights, estimated at 30 per cent. The roofs were very low pitched and could almost be assumed to be flat for area purposes.

The main building and the packing building only require moderate ventilation, but there is a bit of steam in the other building which, for its removal, will call for good ventilation.

Main building—

120 × 45 × 60 = 324,000 cu. ft. at 1 Btu/cu. ft./hr. = 324,000 Btu/hr.

Secondary building—

85 × 35 × 30 = 89,250 „ „ 2 „ „ = 178,500 „

Packing building—

70 × 40 × 20 = 56,000 „ „ 1 „ „ = 56,000 „

558,500 Btu/hr

Walls—

Brick	at 14 Btu/sq. ft./hr.	80 per cent of area.
Glass	„ 28 „ „	20 „ „ „ „
Av.	„ 17 „ „	100 „ „ „ „

		Btu/hr.
(120 × 60 × 2) + (45 × 60 × 2) =	19,800 sq. ft. at 17 =	336,600
(85 × 30 × 2) + (35 × 30 × 1) =	6,150 „ „ 17 =	104,550
(70 × 20 × 2) + (40 × 20 × 1) =	3,600 „ „ 17 =	61,200
		502,350

Roofs—

Slate on boards at 30 Btu/sq. ft./hr.
Glass „ 36 „ „

The lighted roof consists of approximately 30 per cent glass, so that its average heat loss can be taken at 32 Btu/sq. ft./hr.

120 × 45 = 5,400 sq. ft. at 30 =	162,000 Btu/hr.
85 × 35 = 2,975 „ „ „ 30 =	89,250 „
70 × 40 = 2,800 „ „ „ 32 =	89,600 „
	340,850 Btu/hr.

Total space heating requirements

Ventilation	558,500 Btu/hr.
Walls	502,350 „
Roofs	340,850 „
	1,401,700 Btu/hr.

Space heating will be $\frac{1,401,700}{5,623}$ = say 250 Btu/lb. E.S.

647. CONDENSATE. We have these quantities of condensate from the steam from the various users :—

Washings evaporator (Fig. 370) ..	·607 lb.	containing	109 Btu.
Preconcentrator (Fig. 372)	·902 „		163 „
Final evaporator (Fig. 373)	·937 „		169 „
Meal drier (Fig. 374)	2·236 „		402 „
	4·682 lb.	with	843 Btu.

Let us assume that the recovery is 90 per cent. effective on quantity and 74 per cent effective on heat.

Then we can say that the recovery is 4·214 lb. with 624 Btu.

The condensate from the second effects may be contaminated and is therefore not returned for boiler feed but is used for process. The second effect condensate amounts to

	lb.		*Btu*
Washings evaporator (Fig. 370) ..	·586	containing	87
Preconcentrator (Fig. 372)	·696		102
	1·282 lb.	with	189 Btu.

Let us assume a 90 per cent. return with 80 per cent. of the heat and we get 1·153 lb. with 151 Btu.

648. BOILERS AND POWER HOUSE. We must assume a boiler efficiency as there are no accurate figures available. We will guess, by intelligent observation, that 63 per cent. is about the mark.

The coal is pretty good, say		12,000 Btu/lb.
Coal burnt per week		210 tons
Solids extract per week		241 tons
Coal heat ..	$\dfrac{210 \times 2{,}240 \times 12{,}000}{241 \times 2{,}240} =$	10,456 Btu/lb. E.S.
At 63 per cent. boiler efficiency the heat put into the steam is		6,587 Btu/lb. E.S.

In addition to the coal heat put in the boiler there is the heat that comes back in the condensate, say 624 Btu/lb. E.S.

The total steam heat is therefore 6,587 + 624 = 7,211 Btu/lb. E.S.

The factory management assume that the engines are using 35 lb. steam/kWh which they assert is the figure specified by the engine makers. This is extremely unlikely as it would call for an efficiency ratio of 75 per cent. which is hardly possible on such a small machine. We will assume an efficiency ratio of 65 per cent. when the steam consumption becomes 41 lb./kWh.

The hourly output, see Section 640, is 5,623 lb. E.S.

The factory load is 500 kW or $\dfrac{500 \times 3{,}415}{5{,}623} = 304$ Btu/lb. E.S.

The heat in 120 psi saturated steam is 1,193 Btu/lb.

To produce the needed power calls for a flow through the engine of

$$\frac{500 \times 41 \times 1{,}193}{5{,}623} = 4{,}350 \text{ Btu/lb. E.S.}$$

The remaining steam must be blown through a reducing valve to the 8 psi main and will contain (6,587 + 624) − 4,350 = 2,861 Btu/lb. E.S.

649. SUMMARY OF DATA. We can now summarise our data which covers all the most important processes.

		Btu/lb. E.S.
Estimated heat input in boilers	(Sect. 648)	6,587
Heat in returned condensate ..	(Sect. 647)	624
		7,211
Heating raw material	(Sect. 644)	66
Heating process water	(Sect. 645)	345
Thickening washings	(Fig. 370)	722
Preconcentrating extract ..	(Fig. 372)	1,071
Final concentration of extract	(Fig. 373)	1,113
Drying meal	(Fig. 374)	2,655
Space heating	(Sect. 646)	250
Power	(Sect. 648)	304
Unaccounted for		685
		7,211

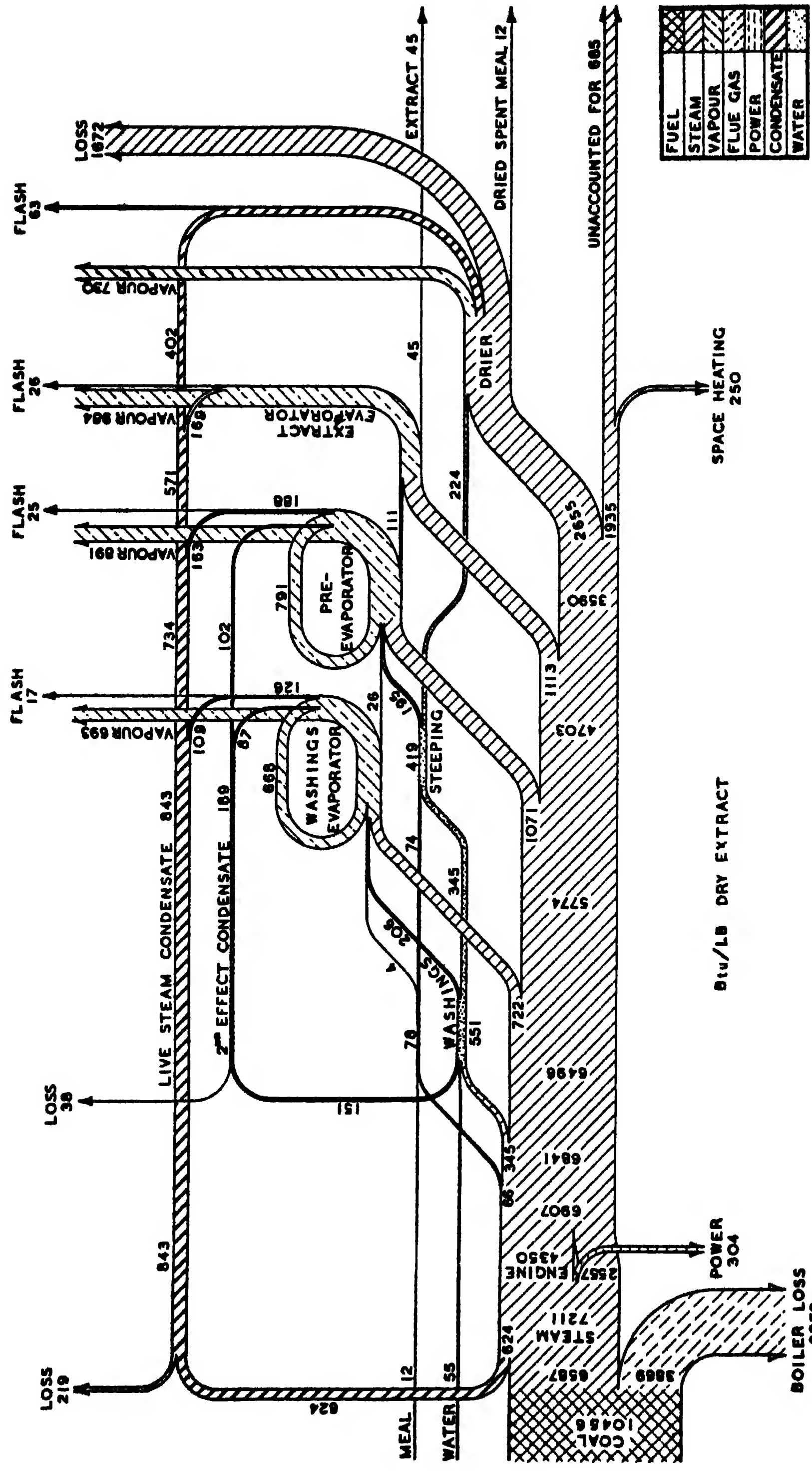


FIG. 375. HEAT DISTRIBUTION IN FOOD EXTRACT FACTORY

This clearly shows that for all practical purposes we have accounted for all the heat, because certain minor processes have been ignored and there are banking and shut down losses that have not been taken into account.

650. CONSTRUCTION OF DIAGRAM. We can put all our information down in the form of a Sankey diagram, Fig. 375.

A little hanky-panky was necessary to avoid complications between the steeping and the preconcentrator. The losses between these processes have been carried forward into the drier.

In view of the admitted roughness of the analysis it comes out remarkably well. As the washing down water, banking and shut down losses, etc., have been ignored, the 685 Btu unaccounted for will almost certainly all be accounted for as soon as a more careful investigation is done. This is one of the very rare first diagrams that does not run true to form. It is superficially excellent.

651. EFFICIENCY OR INEFFICIENCY ? At first sight it might seem that this factory was very good. But is it ? The management knows about multiple effect evaporation, yet there is no vapour heating and no use of flash. Why should not the washings, at any rate, be evaporated in triple or quadruple effect. The extract will probably not be susceptible to high temperature until it is fairly concentrated. Anyhow, by suitably adjusting the heating surfaces it is possible to secure any desired evaporator temperatures in the later effects. It would seem to be very simple and straightforward to use one big triple to do all the evaporation instead of two doubles and a single.

The biggest steam user of all is the meal drier. The spent meal goes for cattle food. Why, then, not copy the beet sugar industry who dry their pulp—also for cattle feed—with flue gases in driers which have efficiencies exceeding 70 per cent ?

Let us take out a diagram on these lines in order to establish a tentative bogey.

652. TRIPLE EFFECT EVAPORATOR. Now we want to do all sensible heating possible with latent heat that has been used as often as possible ; in other words we must heat our raw material input, meal and water, with vapour bled from the evaporator. We will draw Fig. 376 to help us to work out the problem.

We will divide the sensible heating into two separate parts. The extracted solids and all the water used in extraction and process washing as one line ; and the exhausted meal and the water it contains at exhaustion as the other line. Of course in the actual plant this could not be done, several waters would pass through the heaters and the meal, both extracted and exhausted, would take its heat from the water.

We have 8 psi steam at 235° F. and the vapour leaving the third effect at 27 in. will have a temperature of 115° F. But we know from Section 642 that there is a Boiling Point Elevation of 16° F., so that the boiling temperature of the extract in the third body is 131° F.

The overall temperature drop is $235 - 131 = 104°$ F.

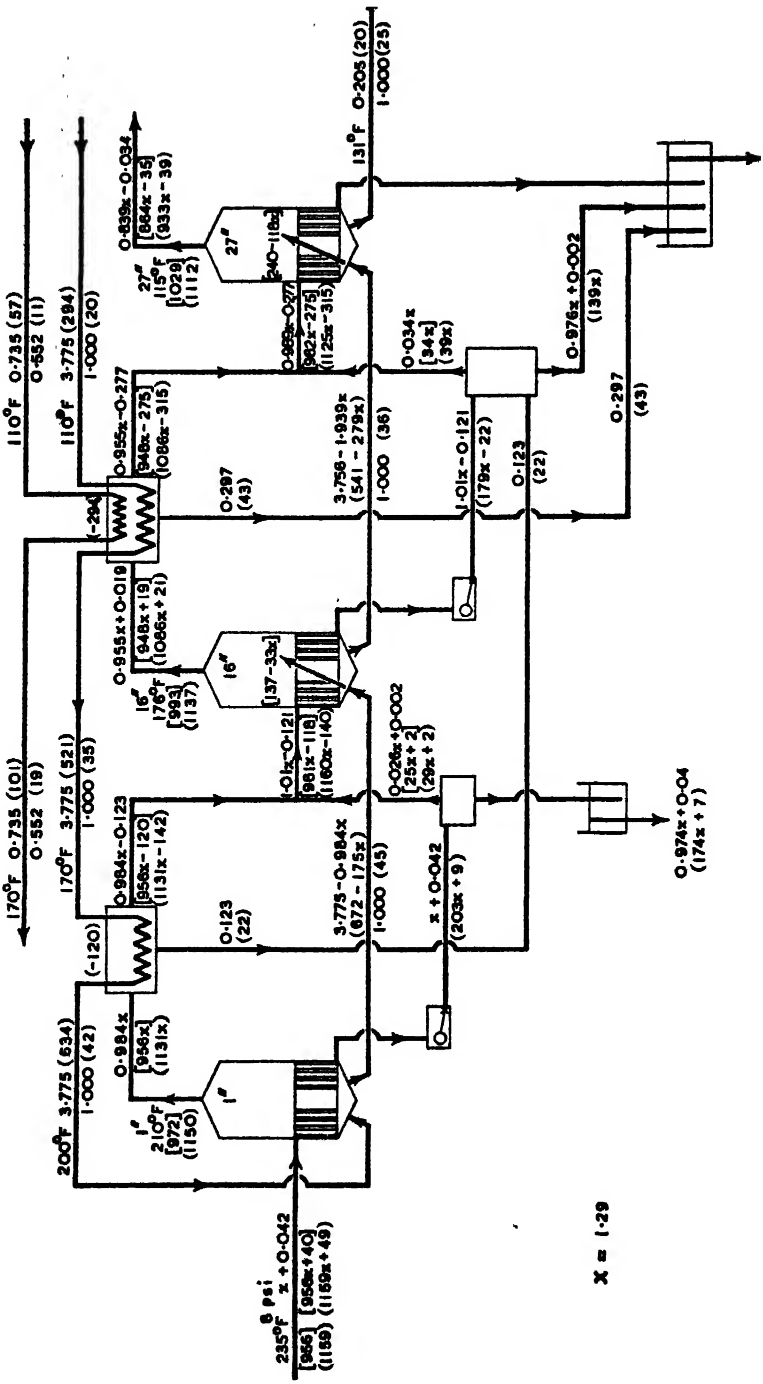


FIG. 376. TRIPLE EFFECT WASHINGS AND EXTRACT EVAPORATOR—ANALYSIS

Suppose we decide upon the following temperature drops over the three effects. We can write the corresponding evaporator pressures alongside.

Evaporator Body.	*Temperature Drop.*	*Pressure or Vacuum.*	*Temperature.*
1st	25° F.	1 in.	210° F.
2nd	34° F.	16 in.	176° F.
3rd	45° F.	27 in.	131° F.

We have ignored B.P.E. in the second body, it will be between 1 and 2 only, and in the first body will be quite negligible.

From Fig. 370 the thin washings consist of	1·495 lb. water and	·113 lb. solids
From Section 641 the thin extract consists of	2·280 " " "	·887 " "
Total feed to triple effect ..	3·775 " " "	1·000 " "
From Section 639 the raw material is	66 tons water and	374 tons solids
For 241 tons of extract solids these figures become	·274 lb. water and	1·552 lb. solids/lb. E.S.
From Fig. 374 the exhausted meal consists of	·735 " " "	·490 " " .

We can ignore the sensible heating of the input meal moisture because it is eventually counted either in the spent meal moisture or in the washings or the thin extract. We will also ignore the actual weight of spent meal solids and will take the difference between the extract solids and the input raw solids for sensible heating purposes. We have, already, found out how much water in process materials has to be heated together with the extract solids. The amount of raw material solids that remain to be heated is therefore

$$1·552 - 1·000 = ·552 \text{ lb.}$$

We thus have two lots of materials to be sensibly heated :—

Thin washings and thin extract ..	3·775 lb. water and	1·000 lb. solids
Raw solids difference and spent meal water content	·735 " " "	·552 " "

Figs. 370, 372 and 373 give the total evaporation to be done as some 3·5 lb. If this is done in triple effect we will recover about two-thirds of it as second and third effect condensates which cannot be used for boiler feed for fear of contamination. Let us assume that we have second effect condensate flashed down to and mixed with third effect condensate of 2·2 lb. at 170° F. We will use this for process, made up with cold water.

The water in the feed will consist of say

2·2 lb. condensate at 170° F. + 1·575 lb. raw water at 50° F.

or 3·775 lb. at about 120° F.

The third effect vapour is at 115° F. and will therefore not be hot enough to do any vapour heating. We will assume that the whole feed enters the second effect vapour heater at 110° F.

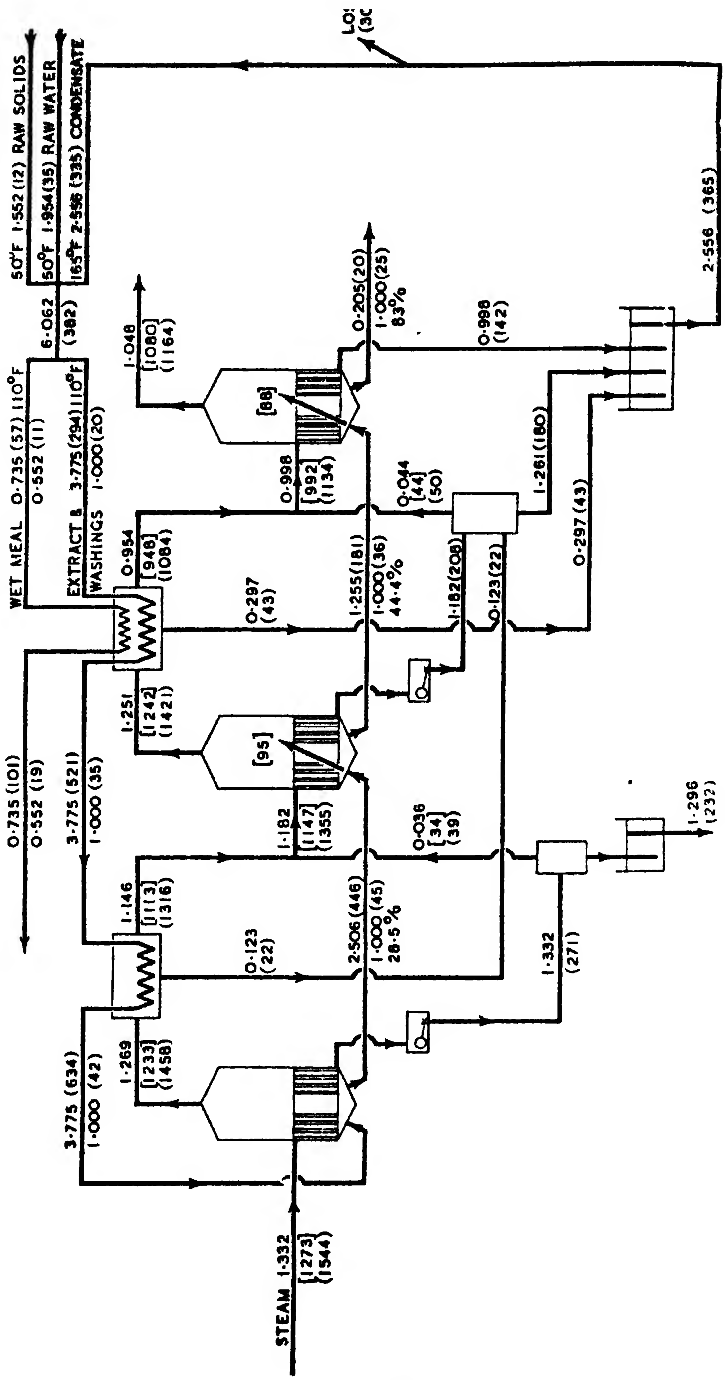


FIG. 377. TRIPLE EFFECT WASHINGS AND EXTRACT EVAPORATOR—SOLUTION

Now the raw material and the spent meal moisture need only be heated to 170° F. and the temperature of the second effect vapour is 176° F. so that this lot of material need only pass through this second heater.

We know the temperatures and pressures of the vapours, so we can write straight in the vapour heating of the process material and the amount of condensate from each vapour heater.

If we assume that the vapour heater on the first effect will heat the feed to 200° F. it will be necessary to heat the feed from 200° to 210° in the first body before evaporation can begin.

This requires $(3{\cdot}775 \times 10) + (1{\cdot}0 \times 10 \times {\cdot}25) = 40$ Btu.

If x lb. of steam are put into the first effect to do evaporation

The latent heat in the input steam must be $956x + 40$ Btu.

The weight of the input steam will be $x + {\cdot}042$ lb.

with a total heat of $1{,}159x + 49$ Btu.

The latent heat available for evaporation is $956x$, and the latent heat at 1-in. vac. is 972.

The amount of evaporation will be $\dfrac{956x}{972} = {\cdot}984x$ lb.

containing ${\cdot}984x \times 1{,}150 = 1{,}131x$ Btu of total heat.

We can now work through the plant as has been explained in Chapter 17.

The evaporation is

$${\cdot}984x + {\cdot}955x + {\cdot}019 + {\cdot}839x - {\cdot}034 = 2{\cdot}778x - {\cdot}015.$$

The evaporation must be $3{\cdot}775 - {\cdot}205 = 3{\cdot}570$

So that $2{\cdot}778x - {\cdot}015 = 3{\cdot}570$

$$x = 1{\cdot}290 \text{ lb.}$$

We can substitute $1{\cdot}29$ for x and produce Fig. 377.

The heat used for heating the raw material and water in Fig. 375 was $345 + 66 = 411$ equivalent to

$$\frac{411}{1{,}159 - 18} = {\cdot}36 \text{ lb. of steam.}$$

Under the original system the steam needed for all three evaporations and for raw material and water heating was

Water and raw material heating ..	·360
Washings evaporator (Fig. 370) ..	·622
Preconcentrator (Fig. 372)	·924
Extract concentrator (Fig. 373) ..	·960
	2·866 lb./lb. E.S.

Fig. 377 shows that with the triple and with vapour heating this has been reduced to 1·332 lb.

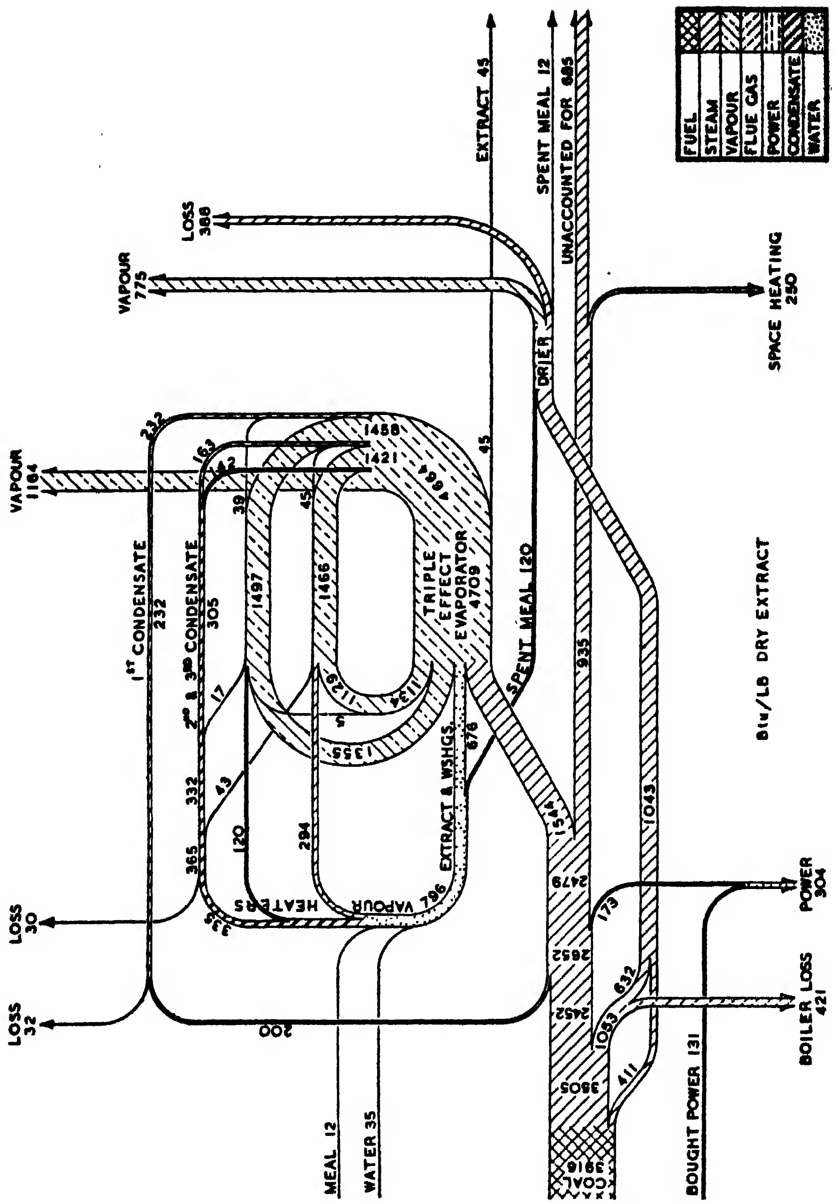


FIG. 378. FOOD EXTRACT BOGEY HEAT DISTRIBUTION

653. MEAL DRIER. In Section 643 it was found that ·674 lb. of moisture had to be removed per lb. E.S.

If only 1·332 lb. of steam is needed for evaporation and process heating, or 1·332 × 1,159 = 1,544 Btu and 250 Btu are needed for space heating, and the same sundries heat, 685, as in Fig. 375 is required, the total steam heat to the process must be 2,479 Btu/lb. E.S.

In Section 648 we took 41 lb. steam per kWh as a probable figure. Now the steam that is available for the engine is much reduced, so that even if we assume that the efficiency of the engine can be screwed up, this gain will be offset by the smaller size of the engine. We will therefore assume that the engine will still use 41 lb./kWh.

The total heat put into the engine in Fig. 375 was 4,654 Btu to turn 304 Btu into power. With the triple effect evaporator we know that the total heat for process need only be 2,479 Btu. If only 2,479 is to be exhausted or lost the engine will only give 173 Btu of power and will call for an input of 2,652 Btu.

From Fig. 377 there is 232 Btu in the uncontaminated condensate from the first effect. If 32 Btu are lost, 200 will be returned as boiler feed. For a very generous bogey we will take 70 per cent. as a reasonable boiler efficiency, so that the coal that must be burnt to give the needed heat is

$$\frac{2{,}652 - 200}{\cdot 7} = 3{,}505 \text{ Btu}$$

and the boiler loss will be

$$3{,}505 - 2{,}652 + 200 = 1{,}053 \text{ ,,}$$

If 60 per cent. of this loss is heat in the flue gas we have only

$1{,}053 \times \cdot 6 = 632$ Btu available for the drier.

We have to do about 730 Btu of drying.

If the drier is 70 per cent. efficient it will call for an input heat of

$$\frac{730}{\cdot 7} = 1{,}043 \text{ Btu.}$$

There is therefore a deficit of 1,043 − 632 = 411 Btu to the drier which must be made up by burning extra coal.

654. BOGEY. We can now construct our Sankey diagram, Fig. 378, for the improved factory, based on the generous boiler efficiency of 70 per cent. We see that we only need to burn coal equivalent to 3,916 Btu/lb. E.S., whereas in Fig. 375 the apparently efficient factory burnt 10,456 Btu—a saving of 62 per cent.

This is a very remarkable result from very simple application of well-tried technique to a factory which at first sight looked very efficient—it could account satisfactorily for 92 per cent. of its heat, a thing that hardly any factory can do.

655. EXTRACT DISCUSSION. If bogey could be achieved, and there is little doubt that it could, because in the foregoing only the two main processes have been considered, the saving would be 6,500 tons of coal a year. At 100*s*. a ton this is some £32,000 a year. The new plant needed is a triple effect and a meal drier. The meal drier must be entirely new, but the triple can well be

made up by adding one vessel to the existing two doubles and single, thus having a triple consisting of three pairs of vessels. Such a plant has great practical convenience because one vessel can be put off for cleaning without sacrificing the whole of one effect.

656. BIG POWER STATION. This example is included for several reasons. First as an example of an efficient plant; second to show how the main process, on which all care in design has been bestowed, can shoulder subsidiary processes into inefficiency; third to show the dangers of thinking in percentages.

The following data was obtained, and has all been based on 100,000 kW base-load running and refers to the more modern part of the station; the older plant is running under the considerable disadvantage of two shift working.

Boiler efficiency: 86 per cent.

Blowdown: Estimated at just over 0·1 per cent. to drain, flash lost to atmosphere.

Make-up: Estimated at 1·1 per cent provided by unit evaporators across the first two bleeds.

Steam—nominal: 650 psi, 800° F.

Turbine: Inlet, 620 psi.g., 790° F.
1st bleed, 130 psi
2nd bleed, 50 psi
3rd bleed, 5 psi
4th bleed, 20 in. vacuum
Exhaust, 29 in. vacuum
Steam consumption, 9·1 lb./kWh.

Generator efficiency: 98 per cent.

Cycle efficiency, Efficiency ratio: Not known.

Output: 100,000 kW.

Triple effect evaporator working with reduced desuperheated boiler steam and discharging into the condenser, to supply additional make-up for the old plant, which has so large a make-up due to its nightly shutting down with consequent heavy superheater blowing, that its evaporators cannot meet its needs. Consumption about 3,600 lb./hr./100,000 kW in the newer machines.

Space heating of offices, stores, laboratory and workshop by steam heated calorifier. Boiler steam reduced to 75 psi and desuperheated. Condensate to main condenser.

Boiler house, turbine room and switch house heated by losses. Particulars of all buildings taken.

The auxiliaries take 4·3 per cent. of the current generated.

657. CYCLE EFFICIENCY. In order to find the cycle efficiency we must find out the heat drop to each bleed point and how much steam each bleed should withdraw. Fig. 379 shows the adiabatic conditions over the various

parts of the turbines. It also shows the adiabatic steam state at each bleed point. These figures were read straight off the Mollier chart.

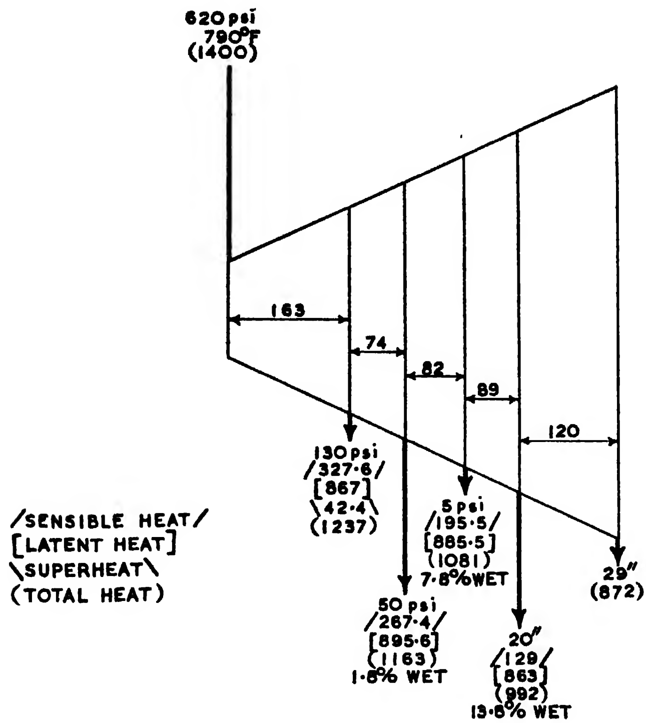


FIG. 379. ADIABATIC HEAT DROPS AND BLEEDS IN LARGE TURBINE

We must find the theoretical quantity of steam that is to be taken from each bleed point. The first bleed feeds the unit evaporator as well as the bleed feed heater, and the evaporator discharges into the second bleed.

We must find out how much steam the evaporator should take to put 1·1 per cent. of make up into the system. The heat in the condensate from the evaporator must be used either for feed heating its own feed, or for heating the main feed—it does not matter which. By passing the condensate into the bleed heater condensate system, a simplification is obtained and a feed heater for the evaporator is saved.

Fig. 380 shows the unit evaporator analysis. 1·1 per cent. of 910,000 lb. is 10,010 lb. If the evaporator blowdown is 2 per cent. the feed to the evaporator must be

$$\frac{10{,}010}{\cdot 98} = 10{,}214 \text{ lb.}$$

To produce 10,010 lb./hr. of make up will call for about 12,719 lb./hr. of steam from the first bleed.

Fig. 381 shows the theoretical performance of each bleed heater. The feed, in thousands of pounds per hour, is passed through from right to left, from heater 4 to heater 1. It is assumed that the water is heated up to within 1° F.

of the saturation temperature of the bleed steam. The condensate is passed from left to right from the higher pressure heaters into the lower pressure. The flash heat from the condensate is thus given up conveniently.

The sensible heat in the feed after each heater can be written straight in. This gives us the heat addition called for in each heater. The last heat addition must be provided by first bleed steam. The condensate from this heater together with the unit evaporator condensate goes into the second heater. The flash heat available must be deducted from the heat requirements of the second heater to find the steam heat needed. The steam output of the unit evaporator must be deducted from this steam to get the actual amount of bleed.

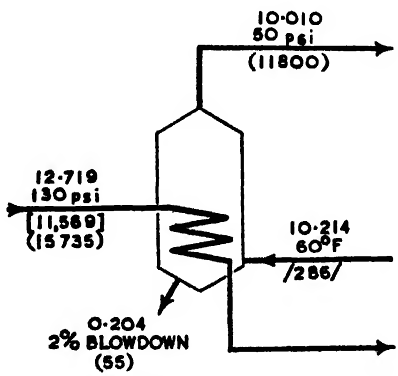


FIG. 380. UNIT EVAPORATOR IN POWER STATION

The combined condensate goes into the third heater, giving up its flash heat, and so on.

We can now tabulate from the information in Figs. 379 and 381 :—

	Pressure	Weight of bleed	*S* Weight of steam in turbine	*D* Adiabatic heat drop	Total heat drop $S \times D$
		lb.	lb.	Btu.	Btu.
Turbine inlet	620 psi	—	910,000	163	148,330,000
1st bleed point	130 ”	73,800	836,200	74	61,878,800
2nd ” ”	50 ”	59,100	777,100	82	63,722,200
3rd ” ”	5 ”	55,300	721,800	89	64,240,200
4th ” ”	20-in. vac.	71,350	650,450	120	78,054,000
					416,225,200

The heat in the steam at the stop valve is .. .. .. 1,400 Btu/lb.

The heat in the feed is 327

The heat supplied is .. .. 1,073 Btu/lb.

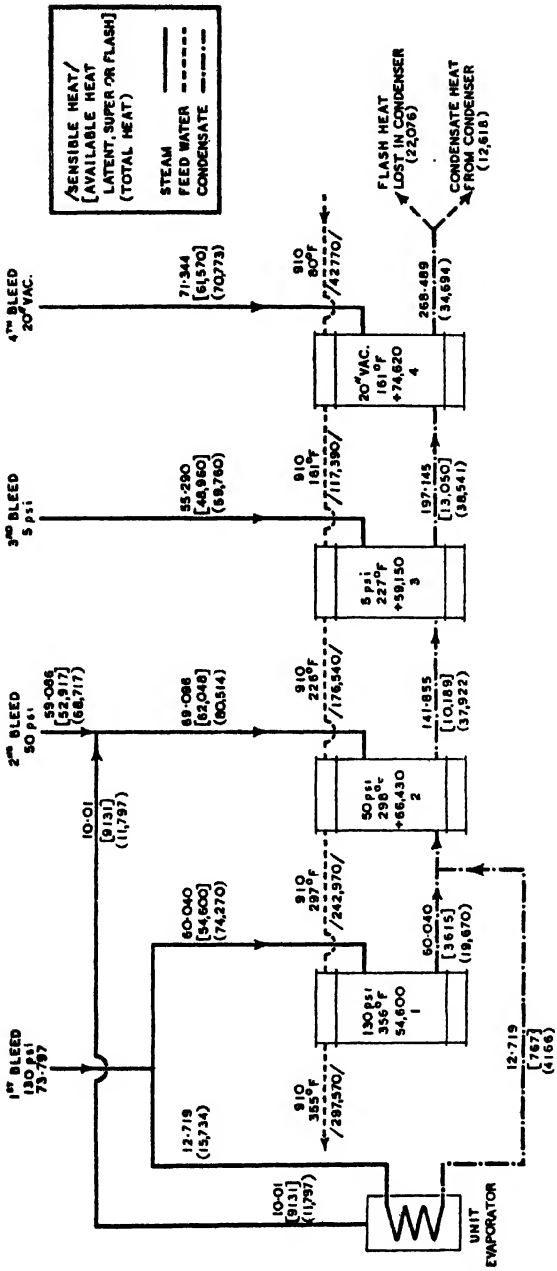


FIG. 381. BLEED FEED HEATER ANALYSIS—ADIABATIC CONDITIONS

The total heat supplied is 910,000 × 1,073	976,430,000 Btu
The total available heat drop is	416,225,200 „
Therefore the cycle efficiency is	42·6 per cent.

658. EFFICIENCY RATIO.

The generator is said to have an efficiency of	98 per cent.
So that, in order to produce	100,000 kW
The turbine must develop	102,041 „
The total heat drop of 416,225,200 Btu is equivalent to ..	121,881 „
The efficiency ratio is	83·72 per cent.

(This figure is so good as to throw some doubt on the data.)

659. OVERALL THERMAL EFFICIENCY. We can now take out the overall thermal efficiency of the boiler-turbo-generator, but not that of the whole station until we have ascertained the other steam users.

The cycle efficiency is	42·6 per cent
The efficiency ratio of the turbine is	83·72 per cent.
Therefore the turbine thermal efficiency is	35·66 per cent.
The generator efficiency is	98 per cent.
So that the turbo-generator thermal efficiency is	34·95 per cent.
The boiler efficiency is said to be	86 per cent.
So that the coal to kilowatts efficiency is	30·06 per cent.
The auxiliaries take 4·3 per cent. of the current, leaving an available output of	95·7 per cent.
Therefore the overall thermal efficiency is approx.	28·8 per cent.

This is a good figure—better than the station actually achieves, but it was based on the steam figure for base load running at economical rating. Actually, the station is running as a base load station, but at overload all day and on about half load at night. Neither of these ratings is efficient and the real efficiency of the station is nearer 27 per cent. than 28·8 per cent. Another cause of reduced efficiency is the rise in summer of the cooling water temperature which prevents the proper vacuum being maintained.

660. BLOWDOWN. If the boiler blowdown is taken at ·11 per cent. it is probably on the low side. This sounds a very small amount, but it amounts to 1,000 lb./hour/100,000 kW. The flash steam lost to atmosphere is over 300 lb./hour, or enough to brew about 200 barrels of beer a week, or to wash 1,000 lb. of laundry a day.

In Section 411 a flash-for-power recovery system with three bleeds was shown in Fig. 236, and some 20 kW was found to be recoverable from 1,000 lb./hour of blowdown at 650 psi.

Fig. 382 shows a similar arrangement applied to the four bleed turbine in the present station. A 1 per cent. loss has been assumed over each flash tank.

This arrangement recovers 380 lb. out of the 1,000 lb. blown down as clean condensate, and adds 23 kW to the generator output.

The plant can be very cheap. If the flash tanks are only 14-in. pipes the steam velocities in the tanks are well below those given in Table LVII. The flash steam velocities are shown in Fig. 382. It is important to keep the velocities reasonably low to reduce the risk of entrainment because blowdown contains the unwanted solids which are to be got rid of, apart from possible damage to turbine blades.

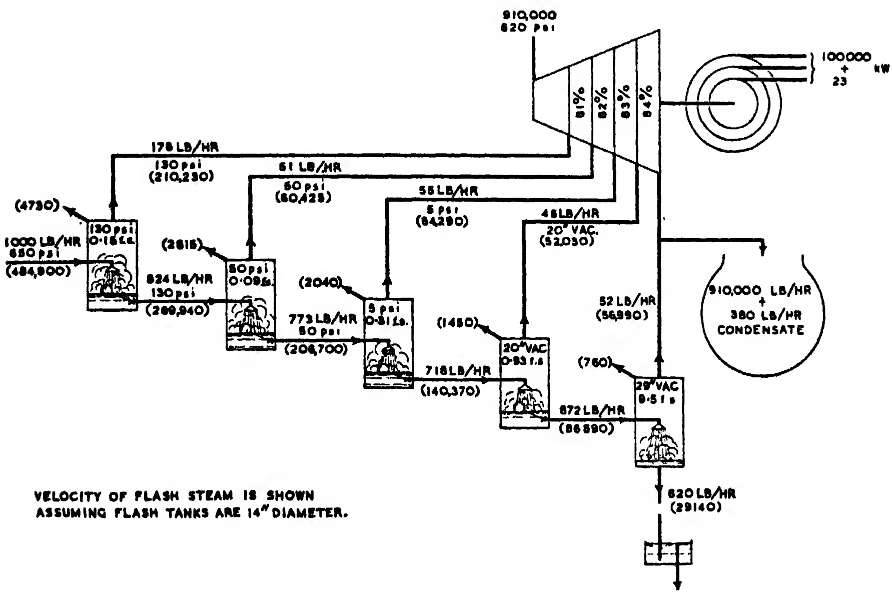


FIG. 382. BLOW-DOWN FLASH INTO BLEED POINTS

661. SPACE HEATING. The calculations of the wall, roof and ventilation losses will not be set down ; they are obtained in the same way as those in the brewery or laundry. The heating requirements seem to be :—

Boiler house, by radiation losses	4,400,000	Btu/hr.	/100,000 kW
Turbine room, by radiation losses	3,000,000	,,	,,
Switch house, by electrical losses	700,000	,,	,,
Office, laboratory, etc., by live steam	2,000,000	,,	,,

The last item is pretty bad. It was done because "it was so simple". It represents only ·2 per cent. of the steam. But what does ·2 per cent. mean ? It is direct loss of either coal or kilowatts or both.

Let us see how much steam is required. The 620 psi steam is reduced to 75 psi and desuperheated.

Each pound of 620 psi steam contains 1,400 Btu

Each pound of 75 psi saturated steam contains 1,186·1 Btu

After desuperheating 1 lb. of 620 psi steam becomes

$$\frac{1400}{1{,}186\cdot 1} = 1\cdot 18 \text{ lb. of 75 psi steam.}$$

One pound of 75 psi steam can give up 895·8 Btu of latent heat.

Therefore the amount of 620 psi steam needed to provide 2,000,000 Btu is

$$\frac{2{,}000{,}000}{895\cdot 8 \times 1\cdot 18} = 1{,}892 \text{ lb./hr.}$$

662. THE EVIL OF DESUPERHEATING. Desuperheating increases the amount of steam and therefore condensate by 18 per cent. This extra condensate carries away its sensible heat to the condenser (in this station) where the sensible heat is reduced from 290·3 to 47 Btu/lb. on 2,233 lb./hour.

If the calorifier were made suitable for use with superheated steam—and why not ?—each pound of reduced and wiredrawn 620 psi steam would give up 1,400 − 290·3 = 1,109·7 Btu. The steam consumption would be reduced to 1,802 lb./hour.

The steam saved, namely 1,892 − 1,802 = 90 lb./hour could go through the turbine (without any extra bleeding because it is already being heated as feed) and could give 11·6 kW and would save the cost of the desuperheater.

663. BLEED SPACE HEATING. Now there is a bleed of 5 psi, which is just exactly right for space heating either via a calorifier or direct as steam. The calorifier is probably better, as a multitude of radiators with low pressure steam are likely to provide much air leakage if the radiators are checked so as to reduce the steam pressure to below atmospheric. Such a calorifier would have to be larger, but not at all exorbitantly large.

Each pound of 5 psi steam can give up 885·5 Btu of latent heat (assuming adiabatic conditions at the bleed point for conservatism).

The heating required is 2,000,000 Btu.

The extra bleed must be 2,257 lb./hr.

We should put into the turbine 1,892 lb./hour of high pressure steam (Section 661) which will be passed out at the third bleed.

This will generate $\dfrac{1{,}892 \times 319 \times \cdot 8}{3{,}415} = 141\cdot 4$ kW.

But this steam is not sufficient at the bleed point and an extra bleed of 2,257 − 1,892 = 365 lb./hour will be needed.

We shall therefore forfeit $\dfrac{365 \times 209 \times \cdot 835}{3{,}415} = 18\cdot 7$ kW.

The net gain will therefore be 141·4 − 18·7 = 122·7 kW.

If we say that the heat is used for 100 hours a week for 28 weeks in the year, the average gain will be about 39 kW.

It is possible that, at night, when the load is light, the 5 psi bleed point will drop in pressure to 10 in. vacuum. It might therefore be necessary to use the 50 psi bleed for space heating during periods of light load.

664. TRIPLE EFFECT EVAPORATOR. The present triple, taking a maximum of about 3,600 lb. of 620 psi steam per hour, is supplying extra make-up to the old plant that its evaporators cannot supply. The plant breaks all the rules of multiple effect evaporation. It operates over a very big temperature drop at a high temperature and does not use its vapour for sensible heating—its vapour heats the local river. There is much loss of heat from condensate flash. The machine is not intended to work continuously. Its real purpose is to fill the station up with distilled water at the original start and to deal with any emergency situation. Even so it could have been designed so as to be an integral part of the plant and possibly to take the place of the unit evaporators which are single effect vessels.

In Sections 470 and 480 it was explained that the multiple effect evaporator works best when the pressures are low so that latent heat which represents the only transferable heat is the largest proportion of the total heat, and when the temperature drops across each vessel are small so that the variations in latent heat from body to body are as small as possible. If, on the other hand, the exhaust and the bleeds from the evaporator can do really useful sensible heating then the advantages of working at higher temperatures are more important than the actual evaporator efficiency. In the case of the power station the steam supply to the evaporator can be taken from one of the bleeds and the exhaust from the evaporator can go into a lower pressure bleed. The lower the pressure of the bled steam the more is power forfeited because the low pressure end of the turbine is more efficient than the high pressure end. On the other hand the lower the temperature of the evaporator the less will be its own heat losses and the less loss by flash heat will there be. We cannot operate the evaporator across the last bleed to the condenser as it is desirable to de-aerate the condensate from the evaporator by some appreciable degree of flash in the condenser. Taking all things into consideration it would seem that the evaporator should take its steam from the last but one bleed and exhaust into the last bleed.

There is no need to plough through the analysis of the suggested triple. We can estimate quite nearly enough from Fig. 288, Section 485. The existing triple is using about 3,600 lb./hr. We do not know its exact performance, but will take the same steam consumption using bleed steam. This may introduce an error, but the actual figures are not so important as the argument.

From Fig. 288 we see that if we put 3,600 lb. of steam in to a triple about 2,000 lb. of third body vapour will be put out, if cold feed is stage-vapour-heated. By putting the triple across the last two bleeds :—

	kW
3,600 lb. steam from 620 to 5 psi with a heat drop of 319 Btu at an efficiency of 80 per cent. will give	269
2,000 lb. steam from 20 in. to 29 in. with a heat drop of 120 at an efficiency of 84 per cent. will give	59
A total maximum power gain of	328

665. SUMMARY OF ECONOMIES. By applying ordinary principles of heat economy technique to the neglected appendages of a big power station, we can make really material savings. They add up to :—

	Average *kW*	*Maximum* *kW*
Blowdown flash recovery	18	23
Bleed space heating	39	122
Triple across last bleeds (say)	100	328
	157	473

Now these totals are very substantial, but they only represent 0·2 to 0·5 per cent. of 100,000 kW. The 160 kW which can be picked up without burning a single extra pound of coal would supply two laundries or one brewery of the size that have already been investigated.

Now this power can be obtained for nothing except the capital cost. If it is not saved the power must be generated in another station where 1 lb. coal per kWh must be burnt. In addition the other power station has to be built and this costs (1957) about £50/kWh. The money savings add up thus :—

	£
Coal per year to produce 157 kW on base load is 610 tons worth £3,000.	
Capitalised at 15 per cent.	20,000
Capital cost to produce 473 kW elsewhere at £50/kW	23,650
	£43,650

Costs are changing greatly nowadays and the following money figures must not be looked upon as proper estimates, but merely as tools to drive home points that are at least worth investigating.

666. ESTIMATE OF COST. The blowdown flash collection consists only of six vessels about 15 in. diameter, five traps and a few hundred feet of small bore piping. It should be possible to do the lot for £5,000.

The space heating by bled steam will probably call for a new and bigger calorifier and a rather more generous steam pipe to carry the lower pressure steam. In a new station the desuperheater would be saved. The work should be possible for about £6,000.

The low pressure triple is rather more difficult to estimate, but some guide can be obtained from experience in the author's factory.

We can say that the cost will probably be about £4,000 on a 1937 basis or say £17,000 in 1957/58.

The total cost becomes :—

	£
Blowdown flash collection	5,000
Bleed space heating	6,000
Low pressure triple	17,000
	28,000
Capital cost saved and capitalised coal saving from Section 665	43,650
Net Bonus	£15,650

Another way of setting out the saving is :—

	£
Cost of alterations	28,000
Capital saving	23,650
Net capital cost	4,350
For an annual gain of	3,000

Or a return of 69 per cent.

This postulates the scrapping of the existing triple, but this plant could still do much of the work across the same temperature difference as the new plant, which could in consequence be smaller and cheaper.

667. POWER STATION DISCUSSION. The foregoing has all been based on the assumption that the turbine can take more steam. Suppose however, as was the fact in the particular station under consideration, that the turbine is flat out. No more steam can be passed into it, so that any extra bleed will reduce its output, not increase it. This gives us a very different cost picture.

The blow-down arrangements can stand because the boilers can perhaps be run during the peak with the blow-down shut, all the blow-down being done during periods of lighter load. The space heating and the triple can only be bled from the steam that is already going through the turbine.

It was shown in Section 663 that the space heating called for 2,257 lb./hr from the third bleed. This will forfeit its heat drop of 209 Btu down to 29 in.—see Fig. 379. The power lost will be $\frac{2{,}257 \times 209 \times \cdot 835}{3{,}415} = 113$ kW.

This is a maximum. For 100 hours a week for 28 weeks it would be 36 kW.

In Section 664 it was assumed that 3,600 lb./hr. would be bled from the third bleed point for the triple effect evaporator. The heat drop between the last two bleeds is 89 Btu. The exhaust from the triple would go into the fourth bleed and undergo a heat drop of 120 Btu to 29 in. The total power loss from the triple will therefore be

$$\frac{(3{,}600 \times 89 \times \cdot 83) + [(3{,}600 - 2{,}000) \times 120 \times \cdot 84]}{3{,}415} = 125 \text{ kW.}$$

If the evaporator works on average 50 hours a week it would be 37 kW.

We thus get :—

	Max.	*Av.*
Power loss from bleed space heating	113	36
Power loss from triple across 3rd and 4th bleed ..	125	37
	238	73
Power gain from blow-down	23	18
Net power loss	215	55

We can legitimately assume that although we cannot put more steam into the high pressure end of the turbine, we can adjust the bleed quantities to a small extent, so that we are justified in taking credit for the blow-down flash recovery.

By taking bled steam for space heating and for the triple we have saved steam but forfeited power when conditions are such that the turbine cannot take more steam.

The steam saving is 3,600 lb./hr. for the triple and 1,892 lb./hr. for space heating.

The boiler feed contains	327	Btu/lb.
The stop valve steam contains ..	1,400	„
Therefore the boiler adds	1,073	„

So that the coal burnt for the triple was :—

$$\frac{3{,}600 \times 1{,}073}{12{,}000 \times \cdot 86} = 375 \text{ lb./hr.}$$

If the triple works for 50 hours a week the coal saving will be :—

$$\frac{375 \times 50 \times 52}{2{,}240} = 435 \text{ tons/year.}$$

The coal burnt for space heating was :—

$$\frac{1{,}892 \times 1{,}073}{12{,}000 \times \cdot 86} = 197 \text{ lb./hr.}$$

In Section 663 is was assumed that the space heating was used for 100 hours a week for 28 weeks, so that the coal saving will be :—

$$\frac{197 \times 100 \times 28}{2{,}240} = 245 \text{ tons/year.}$$

The total coal saving would be :—

$$435 + 245 = 680 \text{ tons/year.}$$

To produce the 55 kW forfeited will require the burning of :—

$$\frac{55 \times 168 \times 52}{2{,}240} = 215 \text{ tons/year.}$$

There will therefore be a net coal saving of :—

$$680 - 215 = 465 \text{ tons/year.}$$

At 100s. a ton this represents an annual saving of £2,325.

	£
Cost of alterations	28,000
Cost of providing 215 kW elsewhere at £50/kW ..	10,750
Total Capital Cost	£38,750

A saving of £2,325 on £38,750 is only a return of 6 per cent. which would not warrant the extra complication, diversion of effort or capital expense.

A turbine costs less per kW to build than the boiler plant. It would seem desirable therefore that stations should be built with spare turbine capacity rather than with spare boiler capacity. But this is hardly possible. A boiler can be pressed beyond its rating, which is a rather indefinite figure being really an output fixed by the boiler maker at as high a figure as he thinks he can get away with. The output of a turbine is a definite figure limited either by the amount of steam that can be pushed through it, or by the temperature of the generator.

The engineer responsible for running a big power station will always resist the introduction of complications, however efficient, especially those which in any way connect two big turbines together. If the space heating and the triple were supplied by bleeds they would have to be taken from at least two machines. This would connect the two turbines together on the downstream side of the governor and stop valve, and all kinds of precautions would have to be taken to prevent the bleed from one turbine driving the other if the other happened to have tripped off the bars. Whereas the brewer is obsessed with the quality of his beer and the sugar refiner with the quality of his sugar, the power station engineer is obsessed with the continuity of supply—this is especially the case in England where the generating industry is still smarting under the stigma of "Black Sunday", 29 July, 1934, when half the electric supply of the country failed. The failure would probably not have occurred had not two of the largest turbines in the country been shut down because it was necessary to repair the one and only valve that connected the two turbines together. Now that the C.E.B. have proper control and the Grid is connected to such huge and scattered capacity, there would seem to be much less need for this extreme caution. Many power stations should therefore be able to effect sizeable economies in their ancillary and auxiliary processes were they to adopt a little of the complication that the process factory takes in its stride.

668. POWER STATION ALTERNATIVE. There is an alternative to drawing extra steam from the bleeds for the auxiliaries. This alternative is entirely free from the objection that it connects the main units together by valves.

The triple requires about 3,600 lb./hr. and space heating calls for about 2,250 lb./hr. If a small back pressure turbo-generator is installed it can take boiler steam and exhaust to these two steam users at about atmospheric pressure. To produce 5,850 lb. steam at exhaust, will, we can guess, call for about 6,000 lb. of 620 psi steam. The adiabatic heat drop will be about 340 Btu and the efficiency ratio 50 per cent.

The output should be

$$\frac{340 \times 6{,}000 \times \cdot 5}{3{,}415} = 300 \text{ kW.}$$

The cost of such a machine, inclusive of piping and switchgear, all of which can be of the simplest, should not (in 1957) exceed about £35/kW.

The present space heating and triple are taking 1,890 + 3,600 = 5,490 lb./hr. The back pressure turbine will take say 6,000 lb./hr. Some 510 lb./hr. extra high pressure steam will be needed, calling for the burning of

$$\frac{510 \times 1{,}073}{12{,}000 \times \cdot 86} = 53 \text{ lb./hr.}$$

or 207 tons a year maximum.

or about 70 tons a year if the average power generated is 100 kW winter and summer.

So we get :—

	Cost	*Power gain kW*	
	£	*Max.*	*Av.*
Blowdown heat recovery	5,000	23	18
New triple	17,000		
Space heating	6,000		
New back pressure turbine, say	10,500	300	100
	38,500	323	118
To instal plant to generate 323 kW elsewhere will call for a capital cost at £50/kW of ..	16,150		
Net Capital Cost	£22,350		

To generate 118 kW elsewhere will require the burning of 460 tons of coal a year whereas with the back pressure set the extra coal burnt will be roughly 70 tons/year.

So that for a capital cost of about £22,000, we can expect a saving of 390 tons of coal a year or about £1,950. This shows a return of 9 per cent. which is a modest reward, but actually the picture is really brighter. We have assumed an entirely new triple. The old triple could still do the bulk of the work and the new plant could be quite small and cheap. The return might well approach 16 per cent, which might be considered worthwhile.

669. AUXILIARIES. Before leaving the power station let us glance at the power-taking auxiliaries as distinct from the steam-taking auxiliaries. The power taken by the auxiliaries must be looked upon as so much sheer waste,

and every possible effort should be made to cut this power demand to the lowest. The auxiliary power load in the station under consideration was :—

	Per cent.	kW
Coal and ash handling gear	·21	210
Fans	1·94	1,940
Boiler feed	1·08	1,080
Condensate extraction	·14	140
Condenser cooling water	·75	750
Light	·13	130
Sundry	·05	50
	4·30	4,300

This is a fearsome penalty to pay for modernity. There is little that can be done about it. Had the boilers been designed to get the best from natural draught, and had the fans been driven by D.C. is it unreasonable to expect a fan power reduction of 20 per cent.—or nearly 400 kW ? Had this been the case a capital expenditure elsewhere of £20,000 and over 1,500 tons of coal a year would have been saved. What would have been the extra cost of a tunnel- or tower-like boiler and D.C. or Ward-Leonard auxiliaries ? Possibly, after deducting £20,000, sufficiently small for £7,500 a year to show a good return ?

What is the efficiency of the boiler feed pump, the other big power eater ? It delivers 910,000 lb./hr. at 825 psi. Table XXXII tells us that 1 psi is equivalent to 2·309 ft. of water. So that the work done by the pump is

$$\frac{910{,}000 \times 825 \times 2{\cdot}309}{2{,}655{,}000} = 650 \text{ kW}$$

The pump is therefore 60 per cent. efficient from cables to delivery pipe, which is very good with a multistage high pressure centrifugal pump.

670. SANKEY DIAGRAM. Fig. 383 is a Sankey diagram for this station. It is not an exact picture, because it is based on the adiabatic bleeds shown in Fig. 381, and it assumes that each feed heater heats the feed to within 1° F. of the bleed saturation temperature. To get the picture exact would have needed many calorimetries of the bleeds, etc., and other figures which were not available. The effect of these approximations is of no importance, because Fig. 383 is only given to show an exceptional occasion where the Sankey diagram is of little value. The main unit, about which nothing can be done—it is either properly designed or it is not—so overshadows the subsidiary processes that, although they are quite large in themselves, they become insignificant threads on the picture.

The subsidiary processes should be given their own diagram on a much larger scale.

It will be seen that each bleed heater on the diagram is shown as rejecting heat to the condenser, which is not shown on Fig. 381. Each of these reject heats represents the ration of loss appropriate to each heater due to the condensate leaving the last heater at 161° F. and leaving the condenser at 80° F.

* * *

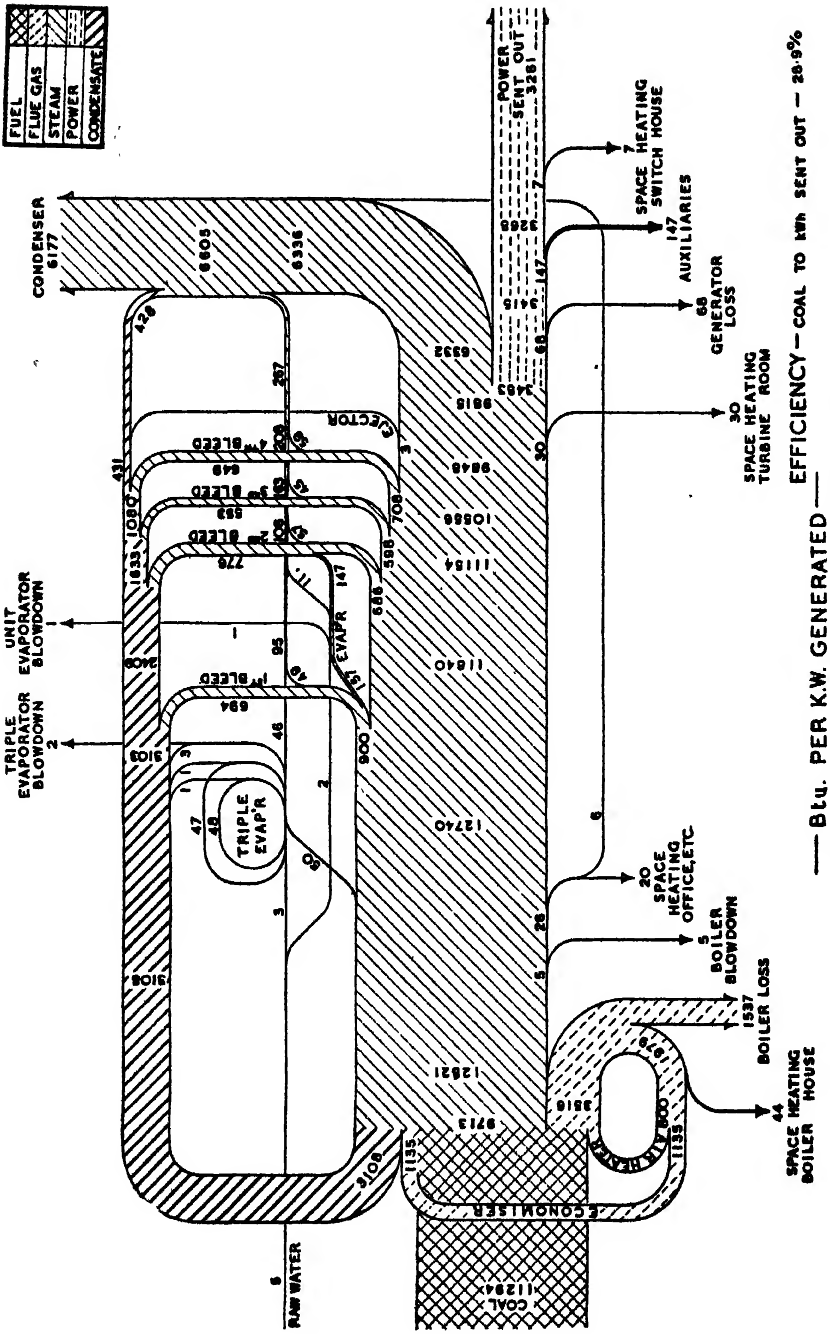


FIG. 383. POWER STATION HEAT DISTRIBUTION

In each of the four different industries whose heat balances have just been considered the method of attacking the problem has been somewhat different. This has been done purposely in order to show possible methods of approach. There is no hard and fast rule. There are always just three things to find out, and it matters little in what order they are investigated, namely :—

How much heat is being used and where is it being used ?

How much heat should be used with the present process ?

How little heat could be used with improved methods or processes?

It is often convenient and desirable to set out the balance or diagram in terms of coal per year or money per week rather than in heat units. It all boils down to money in the end and money rings a bell in the mind of management that remains silent when an attempt is made to impulse it by means of heat units.

CHAPTER 21

COSTING AND REGRESSION

> For money b'ing the common scale
> Of things by measure, weight and tale;
> In all th' affairs of church and state,
> 'Tis both the balance and the weight . . .
> And yet they're far from satisfactory,
> T' establish and keep up your factory.
>
> BUTLER. *Hudibras.* 1663.

THE costing of steam is not a very difficult task. Even if the costing is done in the most elementary way it will give very useful results. But costing alone will not tell us all we want to know. However well we do our steam split (Sections 558 and 559) and however well we cost our steam we can never truly determine the demarcation between fixed steam, that steam used to make up all the process and space heating losses, and the useful or marginal steam, that steam actually used to do useful work in the process. By means of " regression analysis " we can estimate by simple means the fixed and variable heat.

671. SERVICE COSTING. A service consists of something that is produced by material, labour and plant, and which is generally available to any job or process in the works—it is "on tap". The distinction between a service and a material is not always clear. If water or gas, which may be on tap throughout the factory, have no cost added to them over and above their price, there is no need to treat them as services. In the case of the author's factory, water is treated as a service because all the incoming town water is pumped to a considerable height and some cost, pumping, valve maintenance, meter upkeep, etc., is incurred. If water were treated as a material there would be no rational way of allocating these small costs.

Factories may have many different services. Here are some of them :—

Cold water	Hydraulic power
Hot water	Gas
Steam	Oxygen
Electricity	Light
Compressed air	Transport
Vacuum	Laboratory, etc.

The allocation of services presents some of the same problems as the allocation of overheads, but, as in most cases services can be metered or closely estimated, the allocation to departments (if a job costing system is used) or to processes (if a process costing system is practicable) can be done with considerable accuracy.

672. STEAM SERVICE COSTING. Steam is, in very many factories, by far the most important and costly service. If a service such as steam is costed per unit—say per 1,000 lb. of steam—then measurement or good estimate will enable the correct steam cost to be allocated to each part of the factory where steam is used.

If steam is costed at its particular pressure and temperature and is subsequently degraded by passing through an engine, evaporator or reducing valve,

complications at once arise. So steam should be costed " from " the average feed water temperature before any feed heating, " at " 212° F., or dry saturated at atmospheric pressure. Then at the point-of-use, pressure, temperature or quality can be measured and the used steam can be brought back to the common denomination of dry saturated at 212° F.

Another equally good method is to cost the steam per 1,000 Btu or per therm. It is really a matter of personal choice, but the standard steam method has advantages as it is easier to think in pounds of steam than in therms of steam, though this is probably simply a matter of custom.

It is of little use to try to cost steam only. No costing method applied to one process or to one service will be reliable. If steam is to be costed, everything must be costed. The reason is that in any costing system there are items that cannot be measured but have to be allocated. Unless the whole of these items are allocated to something there is no certainty that every expense has been charged to something. An attempt to cost steam alone will almost certainly result in the cost being too low.

On the next page is a Cost Sheet for steam in the author's factory, just before the war (The object of going to three places of decimals is to make certain of getting the first place right. It entails no extra labour as the machine is made for three places.)

This cost sheet is simple and is largely self-explanatory. It gives all the necessary information except one important item, without being burdened with a lot of detail, and the clerical work entailed in getting it out is relatively small. The important exception is the cost of overtime for maintenance. There seems no simple way of showing this satisfactorily in the cost sheet.

It will be seen that water is treated as a service. This refinement gives very little extra trouble and allocates correctly the power and maintenance on the fresh water pumps, valves and meters.

This type of cost sheet is believed to be equally suitable for large or small works and for either steam or hot water. But it only gives the cost per unit of steam. It does not in any way attempt to allocate the steam costs to the process ; that must be done in the heat balance and the steam split.

673. LABOUR. Subdivisions are provided for splitting the labour into small sections, for example there might be two boiler houses, or coal handling might be very awkward and expensive and it might be worth while splitting the labour cost into small operations. For example in 1939 coal and ash handling were taken together, whereas in 1944 with worse coal and more ashes the charges have been separated.

The expense called " Incidentals " is not a general costing term. It is an abbreviation of " Incidental to Wages " and includes all those items other than wages which cease automatically when a man or woman leaves the employ. It includes holiday pay, sick pay, gratuities, National Insurance, employers' liability, overalls and laundry. The item differs from other overheads, which are generally regarded as being proportional to wages but which do not fluctuate exactly with wages ; such items for example as surgery, recreation room,

COST SHEET A

PROCESS—STEAM SERVICE CODE No. 130

UNIT—10,000 LB. STEAM FROM 15° C. AT 100° C.

NET OUTPUT, 2,782 UNITS	ONE WEEK ENDED SEPTEMBER 10, 1939		CUMULATIVE	
			22 WEEKS	23 WEEKS
EXPENSE	£	PENCE/UNIT	PENCE/UNIT	PENCE/UNIT
Labour				
1. Boiler and water softening ..	105·916	9·137	9·487	9·470
2. Coal and ash handling ..	13·162	1·135	1·128	1·129
3.				
4.				
5.				
6.				
7.				
Incidentals	18·048	1·557	1·509	1·511
Overtime	19·716	1·701	·991	1·025
Total Labour	156·842	13·530	13·115	13·135
Material				
1. Coal, 1,107 tons	1,600·452	138·069	144·767	144·449
2. Chemicals	35·016	3·021	3·176	3·169
3.				
Total Materials	1,635·468	141·090	147·943	147·618
Maintenance Allocation				
1. Labour..	73·122	6·308	6·602	6·588
2. Incidentals	9·505	·820	·858	·856
3. Material	91·520	7·895	8·038	8·031
Total Maintenance	174·147	15·023	15·498	15·475
Services				
1. Vac. and hydr. power				
2. Water	58·569	5·053	5·144	5·140
3. Steam				
4. Electricity	—	—	·552	·526
5. Remelting				
6. Inside transport	·896	·077	·091	·090
Total Services	59·465	5·130	5·787	5·756
Direct Cost	2,025·922	174·773	182·343	181·984
Overhead Allocation	58·294	5·029	5·776	5·741
Depreciation	179·426	15·479	16·936	16·866
TOTAL REFINERY COST	2,263·642	195·281	205·055	204·591

canteen subsidy, etc., which, in the author's factory, are included in factory overheads. Overtime is kept separate, thus enabling a good check to be kept on this important and insidious charge.

674. MATERIALS. By far the largest of these is coal. If the water comes straight to the boiler house without having a cost incurred on it, it would be charged as a material. If power for the boiler auxiliaries were bought from the public supply it might be treated as a material.

675. MAINTENANCE. Maintenance is the one item that cannot be really satisfactorily charged. If the actual maintenance costs are used, very irregular results occur whenever such things as the complete renewal of a grate or the annual survey of the boiler take place. The weeks when these things occur give costs that are unduly high, leaving all the other weeks unduly low. It is therefore necessary to take out the maintenance costs but to put in an average cost each week. This average can be that actually incurred in the previous year or a moving average of 2 or 3 years can be used. The last method gives the most satisfactory result and is to be recommended. It may be found that boiler maintenance is so steady that a constant figure is near enough, such a constant figure being adjusted from time to time when any major cost, such as a grate or economiser renewal takes place. It will be seen that maintenance only accounts for about 7 per cent. of the steam cost, so that a 10 per cent. error in the maintenance charge for any one week will affect the cost by less than 1 per cent.

676. SERVICES—GROSS OR NET OUTPUT. The costing of services has already been discussed, and the reasons that call for any particular item to be treated as a service have been argued. There is, however, one obvious difficulty.

When costing a boiler house, one of the costs is the energy required to drive the boiler house auxiliaries. If the power for the auxiliaries is purchased from the grid, there is no difficulty. If however, the power is supplied by electricity produced from the steam to be costed, the steam cannot be costed until the cost of power is known, but the power cannot be costed until the cost of steam is known.

The difficulty can be overcome by means of simultaneous equations, or their equivalent.

Let B = cost of operating the boiler house, excluding auxiliary power.

P = cost of operating the power plant, excluding the steam energy used.

S = units of standardised steam produced.

E = units of power generated.

e = energy absorbed by the power plant expressed as standard steam units on the heat basis. (See Section 679.)

a = power taken by the boiler auxiliaries.

x = cost of one unit of standardised steam.

y = cost of one unit of power.

Then

$$x = \frac{B + a\,y}{S} \quad \dots \quad \text{(i)}$$

$$y = \frac{P + e\,x}{E} \quad \dots \quad \text{(ii)}$$

Consider equation (i). The term $a\,y$ is the cost of running the auxiliaries and is spread over all the steam cost. This is quite legitimate where the steam is used only for heating or only for power. But where back pressure machines are used, any increase of $a\,y$ will almost certainly be due to the adoption of a more elaborate boiler plant in an endeavour to generate more power. Such an increase in the cost of running the boiler auxiliaries should be a charge on power and should not be debited to the steam-using process. Equation (i) therefore does not seem to give the right steam cost.

Consider equation (ii). We can rearrange it thus

$$E\,y = P + e\,x$$

or

$$x = \frac{E\,y - P}{e} \quad \dots \quad \text{(iii)}$$

Combining equations (i) and (iii) we get

$$\frac{E\,y - P}{e} = \frac{B + a\,y}{S}$$

$$S\,E\,y - S\,P = e\,B + e\,a\,y$$

$$y\,(S\,E - e\,a) = e\,B + S\,P$$

$$y = \frac{e\,B + S\,P}{S\,E - e\,a} \quad \dots \quad \text{(iv)}$$

In a back pressure plant e will always be much smaller than S because $S - e$ represents the exhaust steam heat. It is unlikely that e will often be greater than $\cdot 2$S. At moderate pressures $a =$ about $\cdot 1$E.

So that $S\,E - e\,a = S\,E - (\text{about } \cdot 02\ S\,E)$

$= \text{about } \cdot 98\ S\,E.$

Suppose that, in so striving after maximum power production in a very small back pressure plant, the boiler house be made so highfalutin that its auxiliaries absorb almost all the power generated, then

a will nearly $= E$

e might be $=$ about $\cdot 3$ S

$\therefore\ S\,E - e\,a =$ about $\cdot 7$ S E.

If the state of affairs were really such that the boiler auxiliaries took almost the whole of the power, the few net units actually sent out would cost an enormous amount and y should approach infinity. In order that y can approach infinity $S\,E - e\,a$ must approach zero, which, it has been shown, can never happen. So equation (ii) or (iv) must be wrong.

The object of a boiler plant and a power plant is the production of steam and/or power. There is nothing to be gained by costing the boiler *house* or the power *house*. We want to know the cost of the *steam* and the available *power*.

If the power used by the boiler auxiliaries be ignored, the costing becomes much simpler.

Let x' = cost of one unit of steam ignoring auxiliaries.

y' = cost of one unit of net power.

$$x' = \frac{B}{S} \qquad \ldots \qquad (v)$$

$$y' = \frac{P + e\,x'}{E - a} \qquad \ldots \qquad (vi)$$

By equation (v) the external expenditure of the boiler house is divided by the steam output and the steam cost per standard unit varies relatively slightly with the pressure and temperature at which steam is produced. Any increased maintenance or capital charges incurred by the generation of very high pressure steam will fall on the steam cost.

By equation (vi) the higher the boiler plant falutes the smaller $E - a$ becomes, whereas P and e do not vary very much, rising only slightly with increased falution. Were the boiler plant to become so high-brow that a almost $= E$, y approaches infinity and we realize that we have installed a crazy plant.

By using equations (i) and (iv) we can operate a lunatic plant where the boiler house uses all the power we can produce yet the power costs appear low.

By using equations (v) and (vi), that is by costing the net power output and ignoring the boiler auxiliaries, we bring out a most important point in the planning of back pressure plants ; a point that is often overlooked. The more power we try to generate by raising the boiler pressure, the more power is taken by the boiler auxiliaries, and there may come a point where there is no net gain in available power.

By ignoring the auxiliaries and costing the net power output, all the additional power costs of a higher pressure boiler plant fall on power generation, as indeed they should on a balanced back pressure plant.

Now what is sauce for the back-pressure-electrically-driven-auxiliary goose should be sauce for the simple heating-boiler-steam-driven-auxiliary gander.

Let us cost a simple boiler plant on its gross steam output and charge it with the steam used to drive its auxiliaries.

Let B = Cost of operating the boiler plant excluding energy for auxiliaries.

S = Gross units of standardised steam produced.

a = Units of steam used by auxiliaries.

x = Cost of one unit of gross steam output.

Then

$$x = \frac{B + a\,x}{S} \qquad \ldots \qquad (vii)$$

$$S x = B + a x$$
$$S x - a x = B$$
$$x = \frac{B}{S - a} \qquad \text{(viii)}$$

But $S - a$ = net steam output and B is the expenditure excluding auxiliaries. So we see that costing on the gross steam output and debiting the steam driven auxiliaries is the same as costing the net steam output and ignoring the auxiliaries.

The reason why the auxiliaries can be ignored is because their energy consumption circulates round and round the plant. This does not apply to the energy taken from the steam by the power plant.

Some examples have been worked out for a case with back pressure power plant with electrically driven boiler auxiliaries.

First using the method with equations (i) and (iv) where the gross steam output and the gross power output are costed, and where the cost of power to the auxiliaries is debited to the boiler plant.

Second using equations (v) and (vi) where the boiler auxiliaries are not charged to steam, and power is costed on the net output by deducting the power taken by the auxiliaries from the gross power output.

In each of these two cases three examples are taken.

(1)	Process steam requirements	100,000 lb./hr.
	Factory power demand	3,000 kWh.
	Boiler auxiliaries take	300 kWh.
(2)	Process steam requirements	80,000 lb./hr.
	Factory power demand	3,000 kWh.
	Boiler auxiliaries take	750 kWh.
(3)	Process steam requirements	100,000 lb./hr.
	Factory power demand	4,000 kWh.
	Boiler auxiliaries take	750 kWh.

For exemplary simplicity it will be assumed that the boiler house cost B, excluding auxiliaries, varies directly with the steam output and that the power house cost P excluding the energy taken from the steam varies directly with the power output.

It is assumed that standardised steam contains 600 CHU (1,080 Btu).

It is assumed that the power plant takes energy out of the steam equivalent to 10 per cent. more than the electrical equivalent of heat.

It will be seen that by debiting the auxiliaries to steam the effect of elaboration of the boiler plant is to leave the power cost virtually untouched while the cost variations are carried by the steam. Where the boiler auxiliaries are ignored in steam costing, and the power used by the boilers is deducted from the gross power output, the cost of steam remains virtually constant and all the power-using boiler house frills are charged to power. Probably the really correct method is to debit to steam such a power charge as would be needed to operate the auxiliaries of a low pressure boiler plant generating process

steam only. Any extra power used in the boiler house due to the generation of power would not be debited to steam but would be deducted from the power plant output. This is the only way in which it can be seen whether it were better to buy some or all the power, or to generate it.

	Costing Gross Steam Production Costing Gross Power Production Debiting Auxiliaries to Steam Equations (i) *and* (iv)			*Costing Gross Steam Production Costing Net Power Production Ignoring Auxiliaries* Equations (v) *and* (vi)		
	(1)	(2)	(3)	(1)	(2)	(3)
B (pence) .. .. =	4,000	3,338	4,180	4,000	3,338	4,180
P (pence) .. .. =	540	614	777	540	614	777
S ($\times 10^4$ lb.) .. =	11·148	9·304	11·652	11·148	9·304	11·652
E (kWh) .. .. =	3,300	3,750	4,750	3,300	3,750	4,750
e ($\times 10^4$ lb.) (heat basis see Section 679) =	1·148	1·304	1·652	1·148	1·304	1·652
a (kWh) .. .. =	300	750	750	300	750	750
x (d. per unit) .. =	367	383	378	359	359	359
y (d. per kWh) .. =	·291	·295	·295	·317	·361	·343
Total expenditure B + P =	4,540	3,952	4,957	4,540	3,952	4,957
Costed net output $(S - e)x + (E - a)y$ =	4,543	3,949	4,960	4,541	3,955	4,962

In Cost Sheet A on page 686 it will be seen that there is a very small charge for electricity service. This was for power bought from the supply company at week ends. The power used by the auxiliaries and generated by the factory power plant is ignored.

677. OVERHEADS AND DEPRECIATION. These are important charges and should be suitably allocated to any service just as to any other part of the factory operations. No attempt will be made here to suggest how best the allocation of these two items should be done. Everyone has their own ideas, and the allocation of these charges is largely a matter of opinion and varies with the general financing policy and methods of the concern.

678. SHORT-TERM COST COMPARISONS. Costs taken out weekly will show considerable variation, and it is generally not possible to say with certainty that one week, whose figures come out less than those of the previous week, is really better than the other. One of the principal causes of these inaccuracies is in the estimation of stocks ; in the case of steam costs, the estimation of coal stock either in the bunker or on the coal heap. If the costs are taken out over longer intervals they will be much more reliable but a long delay will occur before they are available. They should therefore be taken out at the most frequent possible intervals, say weekly, and the results should be cumulated, either over the current financial year or half-year, or a moving average can be kept. Then by comparing the weekly results with the average, a consistent improvement can be detected even if there are local irregularities. There are other reasons for taking out the costs as frequently as possible—these reasons will be discussed in a minute.

While we found it essential to compare actual heat consumption with a bogey figure, it is equally desirable that actual costs should be compared with a bogey. This bogey is usually called the " Standard Cost ", and it can be obtained in a similar way to any other bogey.

679. CHARGING STEAM TO USERS. In Sections 120, 566 and 567 the debiting of heat to back-pressure and pass-out power has been discussed. For the proper costing of heating or power the proper share of the whole steam cost is required, not just the heat equivalent.

This applies to the ascertainment of the actual expenses that are being incurred, so as to get the true costs for accountancy purposes. If, however, a scheme is being considered for saving steam or power, it can legitimately be argued that the "fixed" costs must be ignored, and that the only costs that can be considered are those that are "marginal"—namely those that vary directly with the consumption. The split between fixed and marginal costs can only be found by regression analysis, which is dealt with in Sections 680–689.

Steam is used either for producing power or for providing heat. If the steam goes only to a power generating machine, or only for heating, no difficulty arises. Steam can be costed at so much per lb., per 1,000 lb., per 10,000 lb. or per ton, and the metered steam is debited to the user.

Complications immediately arise when the same steam is used for power production and then for heating, or for power production in two machines, one exhausting into the other.

It can be argued that, heat and energy being mutually convertible, the straight costing of heat to users is the right method. There is no doubt that this is the only justifiable method in a straight back pressure plant where the power load can easily be met by the back pressure machine. Where pass-out machines are used it may be convenient to cost steam on a power-energy basis, because this method gives a cost figure which can be used to prophesy the effect on cost of an increased or reduced power or steam demand. It does not give the true cost of steam or power, but only the effect of a change of usage when operating part condensing.

The various methods will now be discussed.

Heat basis. The amount of heat put into the steam in the boiler is found, and these heat units are costed either as lbs. of standardized steam (say from and at 212° F.) or as therms. The quality of the steam into and out of the users is found, whence their actual heat consumption is ascertained and debited in suitable proportions.

Adiabatic basis. A sink or final temperature is taken and the adiabatic heat drop from the initial steam state to the final steam state at the sink temperature is debited to the first user. The adiabatic heat drop from the exhaust of the first user (at the initial entropy) to the sink temperature is credited to the first user and debited to the second user, whether it be a power machine or a heating process.

Available energy basis. A reasonably attainable sink or final temperature is taken, and the adiabatic heat drop from the intial steam state to the final sink pressure is debited to the first user. The exhaust of the first user is calorimetered and the adiabatic heat drop from the actual exhaust state (at its actual exhaust entropy) to the sink pressure is credited to the first user and debited to the second user.

Examples—

Suppose we have a colliery winding engine exhausting to atmosphere. The steam supplied to this engine has a certain available power-energy only some of which is used by the engine. If the engine were made to exhaust to a condenser, much more of the available energy would be used and the steam consumption would consequently drop. Another way of making more use of the available power-energy is to exhaust the engine into an exhaust or mixed pressure turbine. Another way of making use of the heat in the exhaust is to use it for a heating process, for example for heating pit-head bath water.

Data—

Assume steam is raised at		180 psi.a.	
at a temperature of		440° F.	
*with a total heat of		1,237·3 Btu/lb.	A
*and an entropy of		1·600	
Assume the engine exhausts at		15 psi.a.	
If the entropy remained at		1·600	
*the total heat in the exhaust would be		1,045·8 Btu/lb.	B
and the adiabatic heat drop	A − B =	191·5 „	C
But suppose the actual heat in the exhaust is found by calorimetry to be		1,122·4 Btu/lb.	D
*with an entropy of		1·714	
The real heat drop is	A − D =	114·9 „	E
Showing an efficiency ratio of		60 per cent.	
If the final sink pressure is		28¾ in. vacuum	
with a temperature of		86° F.	
*At 1·600 entropy the total heat is		871·2 Btu/lb.	F
Adiabatic heat drop from 15 psi is	B − F =	174·6 „	G
*At 1·714 entropy the total heat is		932·8 „	H
Adiabatic heat drop from 15 psi is	D − H =	189·6 „	I
Suppose the actual heat in the turbine exhaust is found by calorimetry to be		989·7 Btu/lb.	J
the real heat drop is	D − J =	132·7 „	K
showing an efficiency ratio of		70 per cent.	

* Read off the Mollier Chart.

Assume steam costs 10*s*. per ton.

The cost allocation based on 1 ton of steam will now be investigated.

Example I.

The winding engine exhaust is sent to the pit-head baths. For simplicity we will assume that the exhaust heat provides exactly the amount of bath heat required and that there is no loss between engine and bath. In order to get our bath heating process exactly comparable with the exhaust turbine in *Example II*, it will be assumed that the incoming cold bath-water is heated to 86° F. by heat exchange with the outgoing bath drain water.

(*a*) *Heat Basis*

Measured heat drop across engine, E .. .. 114·9 Btu
Gross heat in engine exhaust, D .. .. 1,122·4 „
Temperature of input bath water .. .. 86° F.
Available heat to baths, .. 1,122·4 − (86 − 32) = 1,068·4 Btu .. L

Cost to engine .. $\dfrac{114\cdot9}{114\cdot9 + 1{,}068\cdot4} \times 10/- =$ –/11·652

Cost to baths .. $\dfrac{1{,}068\cdot4}{114\cdot9 + 1{,}068\cdot4} \times 10/- =$ 9/0·348

(*b*) *Adiabatic basis*

Adiabatic heat drop across engine, C .. .. 191·5 Btu
Adiabatic heat drop from B to F = G .. 174·6 „

Cost to engine .. $\dfrac{191\cdot5}{191\cdot5 + 174\cdot6} \times 10/- =$ 5/2·770

Cost to baths .. $\dfrac{174\cdot6}{191\cdot5 + 174\cdot6} \times 10/- =$ 4/9·230

(*c*) *Available energy basis*

Energy to winder, A − F − I .. .. .. = 176·5 Btu
Energy to baths, I .. .. = 189·6 „

Cost to engine .. $\dfrac{176\cdot5}{176\cdot5 + 189\cdot6} \times 10/- =$ 4/9·854

Cost to baths .. $\dfrac{189\cdot6}{176\cdot5 + 189\cdot6} \times 10/- =$ 5/2·146

(*d*) *Live steam to baths.*—Suppose the baths had previously been heated with live steam

Bath heat needed, L .. .. 1,068·4 Btu
Available heat in live steam, A−(86 − 32) .. 1,183·3 Btu .. M

Cost to baths .. .. $\dfrac{1{,}068\cdot4}{1{,}183\cdot3} \times 10/- =$ 9/0·348

(*e*) *Free bath heat.*—Suppose there had previously been no baths. The management might legitimately say : " We can heat bath water for nothing, so let's build baths ".

Let us tabulate the foregoing five cases for comparison and discussion.

	Winding cost	*Bath cost*
(*a*) Heat basis	-/11·652	9/0·348
(*b*) Adiabatic basis	5/2·770	4/9·230
(*c*) Available energy basis	4/9·854	5/2·146
(*d*) Live steam for baths	10/-	9/0·348
(*e*) Free bath heat	10/-	—

Now it seems quite legitimate to argue that basis (*e*) is correct. In order to get coal it must be wound, but baths, however desirable, are not essential. On the other hand, a progressive management might well say : "Baths *are* essential. We used to operate basis (*d*). By using exhaust steam we are benefiting both winder and baths. Let us split the saving proportionally over each ".

(*d*) Engine cost	.. 10/–	Bath cost ..	..	9/0·348
Total saving	..			9/0·348
Saving pro rata to (*d*)	4/8·938			4/3·410
Net engine cost	.. 5/3·062	Net bath cost	..	4/8·938

We see that the adiabatic basis (*b*) very nearly effects this split, in this particular case.

Again the Baths Committee might say : " We want hot bath water, but we don't care whether we buy live or exhaust steam—the heat is worth the same to us ". If they bought winder exhaust the proceeds would surely be credited to winding, which is basis (*a*).

It is clear from the foregoing that, in this particular plant at any rate, it is quite impossible to lay down any categorical basis for charging steam cost to users. Each concern must decide the most logical basis to suit local circumstances.

Example II.

Take the same colliery but let the winding engine exhaust into an exhaust turbine.

(*f*) *Heat basis.*

Actual heat drop across engine, E .. .. 114·9 Btu
Actual heat drop across turbine, K .. .. 132·7 ,,

Cost to engine .. $\dfrac{114 \cdot 9}{114 \cdot 9 + 132 \cdot 7} \times 10/- = 4/7 \cdot 686$

Cost to turbine .. $\dfrac{132 \cdot 7}{114 \cdot 9 + 132 \cdot 7} \times 10/- = 5/4 \cdot 314$

(*g*) *Adiabatic basis*—as in (*b*).

(*h*) *Available energy basis*—as in (*c*).

Tabulating for comparison :—

	Winding cost	*Turbine cost*
(*f*) Heat basis	4/7·686	5/4·314
(*g*) Adiabatic basis	5/2·770	4/9·230
(*h*) Available energy basis	4/9·854	5/2·146

It will be seen that in this example there is not a great difference between any two methods. All kinds of arguments can be advanced for or against any

one method. It might however be agreed that basis (f) shows what is actually happening.

Examples III, IV, V and VI.

Assume now that the same boiler plant as hitherto is supplying a process factory which requires power, and heating steam at 15 psi.a. When the power and process loads are in exact balance the conditions are as in *Example I.* We will assume that the factory has a pass-out turbine (we will neglect the steam that must always be passed through the low pressure end to keep it cool) and we will vary the power and process loads up and down separately.

Example III—

Power load reduced by 25 per cent.

State before change as in (a)—

Power load	114·9 Btu
Process load, L..	1,068·4 „

New state—

Power load	86·2 „
Process load	1,068·4 „

Of the process load only three-quarters can now be supplied by exhaust steam, so that live steam must be used to provide

$$1{,}068{\cdot}4 \times {\cdot}25 = 267{\cdot}1 \quad .. \text{ N}$$

Weight of live steam $\quad \dfrac{N}{M} = \dfrac{267{\cdot}1}{1{,}183{\cdot}3} = {\cdot}2256$

Net steam saving is $\quad {\cdot}25 - {\cdot}2256 = {\cdot}0244$

Money saving is $\quad {\cdot}0244 \times 10/- = -/2{\cdot}928$

Now, if we wish to foretell the effect of reducing the power load by 25 per cent., we would surely favour any costing system which shows ¼ of the original power cost equal to –/2·928

	¼ *Power Cost*
(a) Heat basis	–/2·914
(b) Adiabatic basis	1/3·692
(c) Available energy basis	1/2·464

It will be seen that the heat basis gives almost the right answer, while the two energy bases are right off the map.

Example IV—

Process load reduced by 25 per cent.

State before change as in a—

Power load	114·9 Btu
Process load	1068·4 „

New state—

Power load	114·9 „
Process load	801·3 „

Only ¾ of the power can now be generated back pressure. The remainder, $114 \cdot 9 \times \cdot 25 = 28 \cdot 7$, must be produced condensing. The total condensing heat drop is $E + K = 247 \cdot 6$.

∴ 28·7 will call for	$\frac{28 \cdot 7}{247 \cdot 6} = \cdot 1159$ tons	.. O
The total steam saving is	$\cdot 25 - \cdot 1159 = \cdot 1341$ tons	
worth	$\cdot 1341 \times 10/- = 1/4 \cdot 092$	

Let us take ½ of the process cost in the three cases :—

	½ *Process Cost*
(*a*) Heat basis	2/3·088
(*b*) Adiabatic basis	1/2·308
(*c*) Available energy basis	1/3·536

Here we see that the heat basis is out of the running and that the available energy basis gives the nearest figure. The heat basis will give the correct result if the increased power cost is deducted from the apparent process saving.

Example V—

Power load increased by 25 per cent.

As the process requires no extra steam, all the extra power must be produced condensing.

Before the change—as in (*a*)—

Power load	114·9 Btu
Process load	1068·4 „

New conditions—

Power load	143·6 „
Process load	1068·4 „

The additional 28·7 of power will call for an extra steam consumption of ·1159 tons—*see* O—costing

$$\cdot 1159 \times 10/- = 1/1 \cdot 918.$$

	½ *Power Cost*
(*a*) Heat basis	–/2·914
(*b*) Adiabatic basis	1/3·728
(*c*) Available energy basis	1/2·464

Again the available energy basis gives the closest estimate.

Example VI—

Process load increased 25 per cent.

The whole of the extra heat must be supplied by live steam through a reducing valve.

Before the change—as in (*a*)—

Power load	114·9 Btu
Process load	1068·4 „

New conditions—

Power load	114·9 Btu
Process load	1335·5 „

$$\text{Extra steam cost } \frac{1335\cdot5 - 1068\cdot4}{1237\cdot4 - 54} \times 10/- = 2/2\cdot918$$

	$\frac{1}{4}$ *Process Cost*
(*a*) Heat basis	2/3·088
(*b*) Adiabatic basis	1/2·318
(*c*) Available energy basis	1/3·536

In this case the heat basis gives almost the correct estimate, and the two energy methods are right out of it.

Let us collect the conclusions from the foregoing examples and set them against the operating conditions.

		Nearest Costing Method
Example III.	Power reduced 25 per cent. Back pressure generation	Heat basis
Example IV.	Process reduced 25 per cent. Part condensing generation	Available energy
Example V.	Power increased 25 per cent. Part condensing generation	Available energy
Example VI.	Process increased 25 per cent. Back pressure generation	Heat basis
Example II.	Two power machines	Any basis

We see that operating in such a way as to generate without any condensing, by purely back pressure working, the saving or loss in money incurred by a change of load can be approximately obtained by multiplying the units of power or process steam in question by their cost on the heat basis. But where operating pass-out, with part of the power produced condensing, the nearest answer is obtained by using the available energy basis. Before deciding whether to adopt this classification for choosing our costing method, there are certain points to be considered.

When costing on the heat basis all the savings secured by the combined generation of power and the supply of process heat accrue to power generation. On the heat basis the cost of steam remains substantially constant.

When costing on the energy basis the power costs per unit remain roughly constant whatever plant is used, whether condensing, pass-out or back pressure, while any savings resulting from back pressure or pass-out working accrue to the exhaust steam.

Now the choice before a factory is almost always between generating or buying power. It is seldom or never that a factory has the choice of buying or producing steam. It would seem therefore that a costing system should be chosen which will show the savings on power so that its cost compared to bought power can be continually viewed.

By using the energy basis all the power is costed as if it were produced condensing, although much or most of it may have been generated back pressure. But, in a pass-out plant, where a fair proportion of the power is produced condensing, the marginal cost must be the condensing cost, so that the energy basis gives us quickly and easily the figure we want for prophesying the effect of a change of load. The real cost of pass-out power varies very greatly according to the power output and/or the pass-out steam demand.

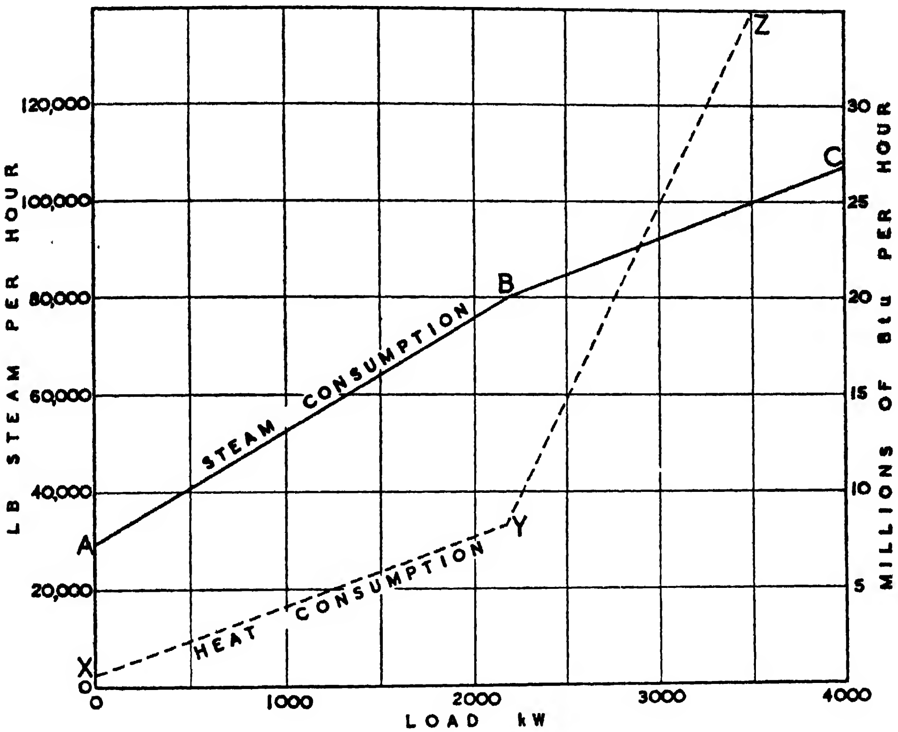


FIG. 384. WILLANS LINES FOR CONDENSING TURBINE PASSING OUT 80,000 LB. STEAM/HOUR

There may often be an essential difference between costing for planning and costing for costs. The matter is important and not at all easy. Fig. 384 gives us a basis for discussion. The diagram shows the Willans Lines of a machine on a lb. steam/power output basis—plain line, and a real heat consumption/power output basis—broken line. The machine considered is a pass-out condensing turbine with a capacity of 3,500 kW. It is assumed that the process demand is constant at 80,000 lb./hour.

When working back-pressure, A B is the Willans line having the formula :—

$$\text{Steam consumption} = 29{,}000 + (23 \cdot 5 \times \text{kW})$$

From B to C the machine operates condensing with an incremental or marginal steam consumption of 15 lb./kWh. Over the back pressure range from A to B the marginal steam consumption is 23·5 lb./kWh.

When working back-pressure the only heat really used by the machine is the electrical equivalent of heat plus the small radiation and gland losses, line X Y. Although the steam consumption is high most of the steam heat goes to process.

When working condensing, from Y to Z, although the steam consumption is much less, the whole of the steam heat must be charged to the machine. The marginal condensing steam consumption is about 64 per cent. of the back pressure steam marginal consumption, but the heat consumption when working condensing is nearly *six times* the back pressure heat consumption ; and for costing it is the heat consumption that we must deal with.

The following tabulation will enable the problem to be more easily studied :—

LOAD	STEAM CONSUMPTION			HEAT CONSUMPTION		
	Total	*Average*	*Marginal*	*Total*	*Average*	*Marginal*
kW	*lb./hr.*	*lb./kWh*	*lb./kWh*	*Btu/hr.*	*Btu/kWh*	*Btu/kWh*
0	29,000			600,000		
500	40,750	81·5	23·5	2,350,000	4,700	3,500
1,000	52,500	52·5		4,100,000	4,100	
1,500	64,250	42·8		5,850,000	3,900	
2,000	76,000	38		7,600,000	3,800	
2,500	85,000	34	15	15,000,000	6,000	20,000
3,000	92,500	30·8		25,000,000	8,333	
3,500	100,000	28·6		35,000,000	10,000	

Suppose the machine is running on a load of 3,000 kW, passing out 80,000 lb./hr. The true cost of heat to each kWh is 8,333 Btu, and this is the figure that must be used for finding the cost of power. But suppose the management are discussing a plant change that will increase the load by 500 kW, they must consider either, that the whole of their power will now call for 10,000 Btu/kWh, or, that the new machine will call for power at 20,000 Btu/kWh. The last method is the simpler and the less likely to lead to error. A decision can then be reached on whether the new machine is justified.

Now comes the rub. The heat cost of all the power will be increased from 8,333 to 10,000 Btu/kWh. Is the new plant to be charged with power at 20,000 Btu/kWh and the old plant at 8,333 ; or is the whole plant to be charged at 10,000 Btu/kWh ? The author favours the latter, because discrimination between one user and another will lead to a mass of complication.

When power costs are taken out on the available energy basis all the power will have been charged at 20,000 Btu. This is quite wrong for costing, but very convenient for planning. On an available energy basis the power cost over line X Y in Fig. 384 would have been charged at 20,000 Btu when in fact the marginal heat is 3,500 Btu/kWh.

When costing on the energy basis it is very likely that a small plant may show the cost of back pressure power equal to that of bought power, while the cost of back pressure steam will appear little more than half the cost of virgin steam. We thus have 1 kWh of grid power costing say 1*d.*, and 1 kWh of back-pressure generated power costing say 1*d.* These are identical as to both energy and cost and should be interchangeable. But the immediate result of

replacing works-generated power by grid power would be to increase the cost of the process steam by about 80 per cent. (in the example on page 695 the steam cost would jump from about 5/- to about 9/-). Any costing system which gives such curious effects is surely open to question. But in spite of this serious objection there are certain advantages possessed by the energy basis.

The energy basis has the property of foretelling the effect of load change in a condensing plant. It also puts a relatively higher value on high pressure steam than on low pressure steam. This is, on the face of it, a good quality, but it is sometimes incorrect. Suppose the process is evaporation that can be adequately done by 10 psi steam. Why should 100 psi steam be valued any higher than 10 psi steam apart from the extra 2½ per cent. of heat that 100 psi steam contains ? It is argued that 100 psi steam could be used in multiple effect where 10 psi steam could not. This is quite true but this is not costing, it is planning. It might indeed happen that high pressure steam was much less suitable than low pressure steam. Suppose the steam is being used for space heating an explosives factory. Owing to the wide dispersal of the plant it does not to pay to return condensate. Therefore only the latent heat of the steam is available. Why should high pressure steam be given a greater value, when, in fact, it contains less available latent heat ?

The energy basis more or less ignores latent heat. Where the use of latent heat is of great importance, as in any heating process, it is essential that the latent heat be costed.

The energy basis possesses another possible danger. Suppose the steam is going into a condensing power plant. Each pound of steam may contain 400 Btu of available energy. The power plant may convert 320 Btu into power and the energy costing system will show an efficiency of 80 per cent. But we know the efficiency of a condensing power machine lies between about 25 per cent. and 35 per cent.

There are pitfalls in costing steam to users on any basis. Provided the coster's eyes are really wide open it probably does not greatly matter which system is used. The ideal system has not yet been published, if indeed it exists. If one factory that requires and generates power but needs no heating, and if another factory, that requires a lot of heating but needs no power, get together and build alongside each other both can greatly benefit. If two such factories agree to operate jointly, the steam-using factory will instal a back pressure plant and will send current to the power-using factory. If the Available Energy system is used the steam will be raised extremely cheaply and the power costs will remain almost the same as in condensing operation. If the Heat Basis is used for costing the power will appear very cheap and the steam-using factory will be charged about the same for exhaust steam as it paid before for raising steam. There seems no simple method of fairly sharing out the savings between the two factories. The author believes that the Heat Basis is the soundest method. It does at least show what is actually happening. But the costs for both power and process steam must always be taken out together in a pass-out plant, and for prophecy the costs must be worked out for many different loads and pass-out steam demands.

The three methods of costing power can easily be compared by referring to the elementary Mollier Diagram, Fig. 384A.

Adiabatic Basis—

A — B = Adiabatic heat drop from 180 psi.a. to 15 psi.a.
A — C = „ „ „ „ 180 „ „ sink.
A — B is charged to first machine.
B — C „ „ „ second machine or process.

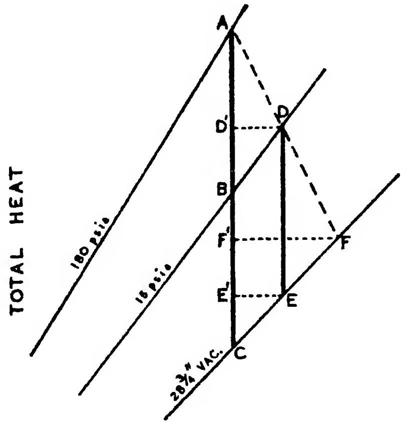


FIG. 384A.

Heat Basis—

A — D' = Real heat drop from 180 psi.a. to 15 psi.a.
D' — F' = „ „ „ „ 15 „ „ sink.
A — D' is charged to first machine.
D' — F' „ „ „ second machine, or
D' — (Sensible Heat at sink temperature) is charged to process.

Available Energy Basis—

A — C = Available energy in 180 psi.a. steam.
D — E = „ „ „ 15 „ „
A — C — (D — E) is charged to first machine.
D — E is charged to second machine or process.

Now D — E = D' — E'

so that A — C — (D — E) = (A — D') + (E' — C)

But A — D' = Real heat drop across first machine.

and E' — C = Loss of energy caused by entropy increase over first machine.

Having decided on the basis upon which steam is to be charged to power generation, it may be possible to debit the steam correctly to power without relying on steam meter or regular calorimetries.

Returning from this long digression on power costing, we will now consider an actual power costing case, using the heat basis.

It is often possible to find one equation which will give the cost of power at varying loads with considerable accuracy.

In Section 120 the turbine in the author's factory was shown to be using

$$71{,}700 \times 123 \quad = 8{,}819{,}100 \text{ Btu/hr. on the heat basis}$$

for a load of 2,280 kW, equivalent to

$$2{,}280 \times 3{,}415 \quad = \quad 7{,}786{,}200 \text{ Btu/hr.}$$
$$\therefore \text{ Loss} \qquad 1{,}032{,}900 \text{ Btu/hr.}$$

The Willans equation (see Section 121), if the losses are constant, must therefore be :—

Steam heat needed $= (1{,}032{,}900 \times \text{hours}) + (3{,}415 \times \text{kWh})$ Btu.

As the losses will increase slightly as the load goes up, due to higher pressures and temperatures causing larger leaks and greater radiation, it is probably fair to cook the equation thus :—

At 2,280 *kW* *At all loads* (*S*)

$$(1{,}032{,}900 \times \text{hours}) + (3{,}415 \times \text{kWh}) = (839{,}000 \times \text{hours}) + (3{,}500 \times \text{kWh}) \text{ Btu}$$

If several calorimetries and steam/power measurements can be taken at different loads the exact equation can easily be obtained by plotting the Willans line.

It is not possible in the author's factory to take tests at different loads because the day load fluctuates too greatly to allow good figures to be got and the steady night load is pretty constant for months at a time.

Steam in this factory is costed "from 60° F. at 212° F.", so that 1 lb. of costed steam contains 1,123 Btu.

The costed steam to be charged to electricity is therefore obtained by dividing the expression (*S*) above by 1,123 which gives :—

Steam for electricity generation $= (747 \times \text{hours}) + (3{\cdot}117 \times \text{kWh})$ lb.

This equation, once obtained, can be used for a long period and need only be changed if the conditions change, such as the overhaul of a turbine or a large change in average load. Occasional checks, twice a year or so, should be done to ensure that leaks or radiation have not increased unnoticed.

Cost Sheet B, on page 704, shows the costs of electricity generation in the author's factory for the same period as the steam costs shown in Cost sheet A, on page 686. The steam cost is related to power generation on the heat basis as this back pressure plant is in power/steam balance.

It will be noticed in Cost Sheet B that the single week shows markedly cheaper cost than the five-month period ; and in Cost Sheet A no electricity is charged to steam in the single week. The reason for both these things is that the factory worked through the week-end of the single week without shutting down. This naturally caused an increase in overtime labour, but the total costs are below normal due to the continuous run.

As already explained the costing of pass-out electricity is more difficult than straight back pressure. The proportions of power to be credited to pass-out and to exhaust may be extremely difficult to assess. It is probably necessary

COST SHEET B

PROCESS—ELECTRICITY GENERATION **CODE NO. 141**

UNIT—1,000 KWH

NET OUTPUT, 437 UNITS	ONE WEEK ENDED SEPTEMBER 10, 1939		CUMULATIVE	
			22 WEEKS	23 WEEKS
EXPENSE	£	PENCE/UNIT	PENCE/UNIT	PENCE/UNIT
Labour				
1.	21·000	11·533	11·070	11·087
2.				
3.				
4.				
5.				
6.				
7.				
Incidentals	3·380	1·856	1·736	1·739
Overtime	5·004	2·748	2·279	2·298
Total Labour	29·384	16·137	15·085	15·124
Material				
1.				
2.				
3.				
Total Material				
Maintenance Allocation				
1. Labour..	17·652	9·694	9·757	9·750
2. Incidentals	2·294	1·260	1·268	1·267
3. Material	8·939	4·909	5·244	5·227
Total Maintenance	28·885	15·863	16·269	16·244
Services				
1. Vac. and hydr. power				
2. Water				
3. Steam	153·119	84·093	87·290	87·069
4. Electricity				
5. Remelting				
6. Inside transport				
Total Services	153·119	84·093	87·290	87·069
Direct Cost	211·388	116·093	118·644	118·437
Overhead Allocation	10·921	5·998	6·646	6·614
Depreciation	40·484	22·234	22·814	22·778
TOTAL REFINERY COST	262·793	144·325	148·104	147·829

to calculate the steam to be debited to power every week in the way shown in Section 567. There it was shown that, in the example given, with 100,000 lb. steam/hour put into the turbine and 20,000 lb./hour passed out, the net steam to be debited to the 7,500 kW generated was 81,838 lb./hour, or 10·89 lb./kWh.

If, in that factory, steam were costed " from and at 212° F." the actual steam debit must be converted to costed steam by multiplying by 1,197, the heat added in the boiler, and dividing by 971, the latent heat at 212° F. The costed steam to be charged to power would then be

$$\frac{10\cdot 89 \times 1{,}197}{971} = 13\cdot 42 \text{ lb./kWh.}$$

It has been so far assumed that it is possible to get a true average sample of exhaust steam and do an accurate calorimetry. If the exhaust is very wet this may be unreliable. In cases where the measurement of exhaust quality is untrustworthy the steam heat to be debited to power should be estimated from Table XIV in Section 120, if the information cannot be obtained from the machine builders.

Where there is no process heating but where there are back pressure machines exhausting to other turbines, the amount of steam charged to each machine should probably be based on the actual heat drop, though the use of the adiabatic heat drop will in such cases not introduce any very large error—the error may be no larger than that introduced in sampling and calorimetering wet exhaust steam.

680. VARIATION OF HEAT CONSUMPTION WITH OUTPUT. How many managers have been told by their staff that heavy steam or coal consumption was due to low output ? How is it possible for management to judge whether this is an excuse or a reason ? Simple statistical analysis will generally go a long way towards providing the answer.

The heat consumption of most factories varies with output. To compare one week with another it is usual to express the steam or coal consumption as a figure per unit of output—so many lb. of steam per lb. output, so many lb. coal per gallon of output, etc. Even so, comparison is not possible if the output varies considerably because consumption per unit of output also varies with the output. The usual relationship between the two is for the heat consumption per unit of output to fall with a rising output.

The staff have no means of judging whether one week is an improvement on another except by searching through past records to find a number of weeks with the same output. This invariably results in an optimistic choice of previous weeks and to wrong conclusions being drawn.

By plotting coal or heat or steam against output a series of points will result, and these points will be more or less scattered—see Fig. 385A. Such a chart is called a scatter diagram and consists of the total weekly output plotted against the total weekly steam consumption.

Fig. 385A is the steam consumption in the author's factory plotted against output every week during the period October, 1936 to March, 1937. A number of very interesting things can be seen by examining this diagram. The scatter is considerable, but there is a very pronounced trend from low left to high right.

We will consider the degree of scatter first. There were six weeks when the output was roughly 7,800 tons and the steam used varied between 32,700,000 and 37,500,000 lb., a variation about the mean of 7 per cent. There were three weeks when the steam consumption amounted to roughly 32,600,000 lb. when the output varied between 6,200 tons and 7,840 tons, a variation about the mean of 11 per cent. This shows how dangerous it may be to make comparisons of individual pairs of weeks.

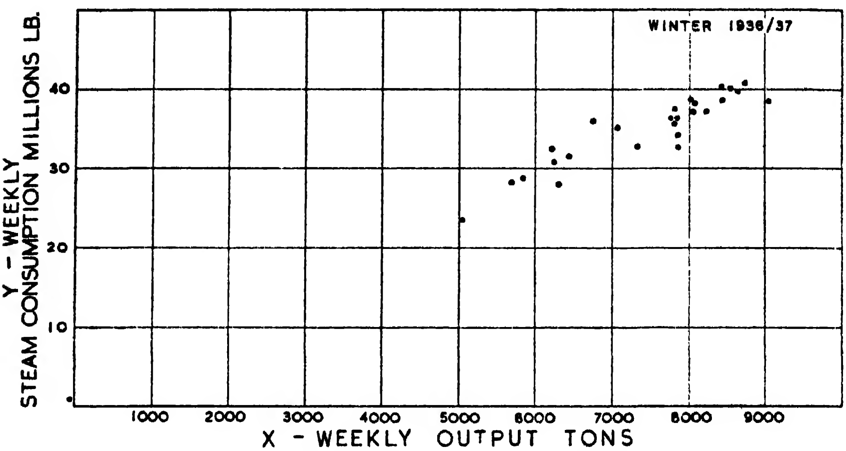


FIG. 385A. STEAM CONSUMPTION/OUTPUT SCATTER DIAGRAM

Now the trend of the points on the diagram shows that there is apparently a distinct relation between output and steam consumption and this relation could be found if we were to draw a straight line such that it passed truly through the scatter of points. If an attempt is made to draw such a line by eye no two persons would draw the same line. By the use of mathematical statistics the proper position and angle of this line can be calculated. Fig. 385B gives the line that calculation shows to be the line of best fit and it is found by using the mathematical trick called the " method of least squares ". No description of the theory of the calculation will be given here, only the actual arithmetical and algebraical method. Those who want to go more deeply into this most useful and interesting control tool should consult " Regression Analysis of Production Costs and Factory Operations " by the author's brother, Philip Lyle, published by Oliver & Boyd, Edinburgh.

An examination of the line of trend of steam consumption with output, called by the statisticians the " Regression Line ", shows that it has its origin at zero output and a steam consumption of about 6,000,000 lb. This is the "fixed" steam that is used regardless of output and is the heat needed to make up for radiation from those pieces of plant which are in operation regardless of the output ; the heat needed for heating buildings ; the heat lost at week-ends, during the start-up and shut-down and so on.

At zero output the steam consumption was about 6,000,000 lb. At an output of 10,000 tons the regression line cuts the steam consumption scale at about 45,000,000 lb. So that the useful, or " marginal " steam needed to turn out 10,000 tons was 45,000,000 − 6,000,000 = 39,000,000 lb., or 3,900 lb./ton of output. Now we have read these figures off the diagram assuming that some superman had drawn in the line at the correct angle and at the correct height. Actually the formula for this line can be got by some simple arithmetic, and the formula is

Estimated steam consumption in pounds per week = 6,246,197 + (3,852 × output in tons)

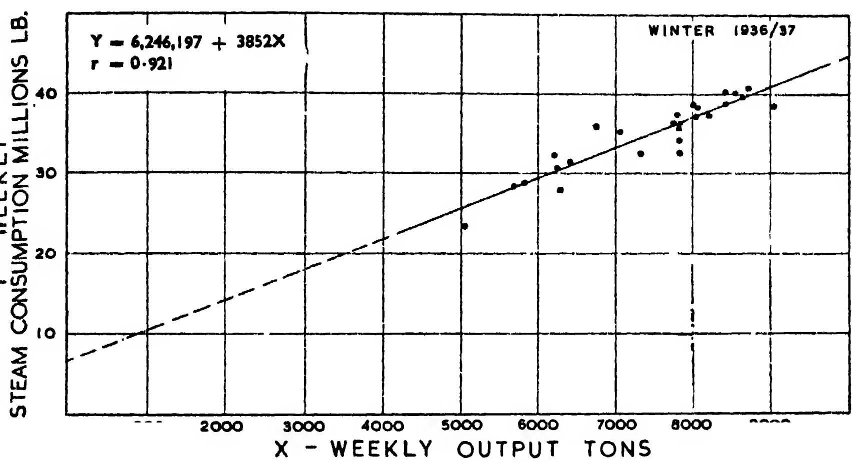


FIG. 385B. REGRESSION LINE

681. THE REGRESSION EQUATION. We will confine ourselves to scatters which can obviously be fitted by a straight line. If it is clear that the line must be curved, more complicated methods are required. A straight line such as we are considering must have a formula similar to that given and which can be expressed algebraically as

$$Y' = a + bX$$

Where Y′ is the total weekly steam consumption in lb. (or coal or therms) as estimated by the regression equation.

a is the estimated fixed steam consumption that is used regardless of output.

b is the marginal steam consumption per ton of output (or regression coefficient)—that is the amount of steam needed to produce an extra ton of output.

X is the weekly output in tons.

The constants a and b which we want to find are given by the following equations :—

$$b = \frac{Syx}{Sx^2} \qquad Syx = SYX - \frac{SYSX}{N}$$

$$a = \overline{Y} - b\overline{X} \qquad Sx^2 = SX^2 - \frac{(SX)^2}{N}$$

Where X is the output in any week.

Y is the steam consumption in that week.

$\overline{X}$ is the arithmetic mean of all the weekly outputs.

$\overline{Y}$ is the arithmetic mean of all the weekly steam consumptions.

N is the number of weeks.

S is an " operator " and means " the sum of ". So that SY means the sum of all the Y's. SYX means the sum of the products of all the pairs of X and Y. SX^2 means the sum of all the squares of the individual X's. $(SX)^2$ means the square of the sum of all the X's.

These equations must be taken on trust—no attempt will be made here to explain, derive or justify them.

Write down all the weekly outputs and the corresponding weekly steam consumptions as set out on page 709.

Add up the two columns. This gives SX and SY.

Divide these sums, SX and SY, by the number of weeks N. This gives $\overline{X}$ and $\overline{Y}$.

SX^2 is the sum of all the squares of the individual outputs or Xs. These squares can either be found from a table of squares or, preferably, taken out on a calculating machine. If a machine is used each square can be left on the machine without clearing and the sum of the squares will be found at the end.

SYX is the sum of all the products of the pairs of X and Y. Again, if these are left on the machine they will accumulate to the sum.

Use as many significant figures as the size of the machine will permit.

The calculation on page 709 shows that our estimate of the steam consumption will have a fixed weekly consumption of 6,246,197 lb. and a marginal consumption of 3,852 lb. per ton of output.

682. CORRELATION. The regression line found by the foregoing statistical analysis is drawn through the scatter of points in Fig. 385B. As far as the eye can see there does appear to be a definite relation between steam consumption and output, and the regression line looks well enough. There is, however, a possibility that the apparent relation may be due to chance and not to any real relation. Even if there is a relation its practical reliability must depend upon the degree of scatter. If the scatter were very much greater than that shown in Figs. 385 we could still calculate the regression equation but its value as an indication of the true relation would be much less.

X Weekly Output Tons	*Y* Weekly Steam Consumption 10,000 *lb.*
8,219	3,731
8,084	3,837
8,653	3,985
8,421	4,026
8,546	4,009
8,732	4,071
9,047	3,853
8,428	3,887
7,814	3,580
7,825	3,629
8,048	3,713
5,064	2,348
6,304	2,791
7,851	3,415
7,323	3,276
7,853	3,259
7,785	3,652
7,073	3,526
7,801	3,744
6,423	3,155
5,834	2,885
6,211	3,244
6,717	3,603
8,039	3,887
5,699	2,802
6,264	3,090
N = 26	194,058 = SX	90,998 = SY
7463·7692 = $\bar{X}$	3499·9231 = $\bar{Y}$

$$b = \frac{Sxy}{Sx^2}$$

$$Syx = SYX - \frac{SYSX}{N}$$

$$Sx^2 = SX^2 - \frac{(SX)^2}{N}$$

$$SYX = 690{,}222{,}009$$

$$SYSX = 17{,}658{,}889{,}884$$

$$SYSX/N = 679{,}188{,}072$$

$$\therefore Syx = 11{,}033{,}937$$

$$SX^2 = 1{,}477{,}046{,}244$$

$$(SX)^2 = 37{,}658{,}507{,}364$$

$$(SX)^2/N = 1{,}448{,}404{,}129$$

$$\therefore Sx^2 = 28{,}642{,}115$$

$$\therefore b = \cdot 38523471$$

$$a = \bar{Y} - b\bar{X}$$

$$b\bar{X} = 2875 \cdot 3034$$

$$\therefore a = 624 \cdot 6197$$

As the Y figures are in units of 10,000 lb. the equation must be multiplied by 10,000.

So that the regression equation relating steam consumption to output is

$$Y' = 6{,}246{,}197 + 3852X$$

The worth or value that we can assign to a regression equation can be checked by finding what is called the "Correlation Coefficient" and testing its so-called "Significance". The correlation coefficient is a fraction the square of which indicates the proportion of true relation that the equation represents, the remainder being due to chance or error.

The correlation coefficient is represented by the letter r and is found from the following formula :—

$$r^2 = \frac{(Syx)^2}{Sy^2\ Sx^2}$$

We have already calculated Syx and Sx^2

$$Sy^2 = SY^2 - \frac{(SY)^2}{N}$$

$$\begin{aligned} Syx &= 11{,}033{,}937 \\ (Syx)^2 &= 121{,}747{,}765{,}719{,}969 \\ Sx^2 &= 28{,}642{,}115 \\ SY^2 &= 323{,}499{,}792 \\ (SY)^2 &= 8{,}280{,}636{,}004 \\ (SY)^2/N &= 318{,}486{,}000 \\ \therefore Sy^2 &= 5{,}013{,}792 \\ \text{and} \quad r^2 &= \cdot 8477926 \\ r &= \cdot 9208 \end{aligned}$$

(The method of taking out square roots on a calculating machine can be obtained from the makers of or agents for the machine.)

Now $r^2 = \cdot 85$ indicates that of the total variation in steam consumption 85 per cent. is due to change in output and 15 per cent. to other causes, including error.

683. SIGNIFICANCE. Now there is a difficulty. Suppose we had only two points. The regression line would pass through both and the correlation would be 1·0, that is to say the correlation would be complete. But clearly a statistical investigation of two points is worthless. We have seen in Section 680 how in problems such as we are considering any two points are likely to have such chance variations as to make any comparison between them unreliable. Suppose, on the other hand, all our 26 points lay on a straight line. The odds against this being due to chance are so enormous that we should be justified in saying that the correlation was perfect and that the equation did exactly represent the relation between steam consumption and output. Clearly the fewer the points the less "significant" is the apparent correlation between the regression equation and the actual happenings. Whereas with very many points we can take the equation relation as being reliable although the correlation coefficient is well below unity.

Table LXXI is adapted from Fisher & Yates' "Statistical Tables" and shows the minimum value of the correlation coefficient for different numbers of points such that the odds are 100 to 1 against the result being due to chance.

The regression line in Fig. 385B was found to be correlated to the scattered points to the extent of ·921. The minimum value of the correlation coefficient

for 25 points for the odds to be 100 to 1 against the relation being due to chance is ·506. So that we can say with confidence that the apparent relation is almost certainly not due to chance.

TABLE LXXI. CORRELATION SIGNIFICANCE

NUMBER OF POINTS	MINIMUM VALUE OF CORRELATION COEFFICIENT SUCH THAT ODDS ARE 100 TO 1 AGAINST IT BEING DUE TO CHANCE
N	*r*
10	·767
15	·641
20	·561
25	·506
30	·464
35	·425
40	·402
45	·380
50	·362

684. WHAT REGRESSION TELLS US. Figs. 385A and B show the results of steam consumption compared to output in 26 winter weeks just before the big steam saving campaign got going in the author's factory. It shows that the fixed steam amounted to 6,246,205 lb./week and that the marginal steam consumption was 3,852 lb./ton of output.

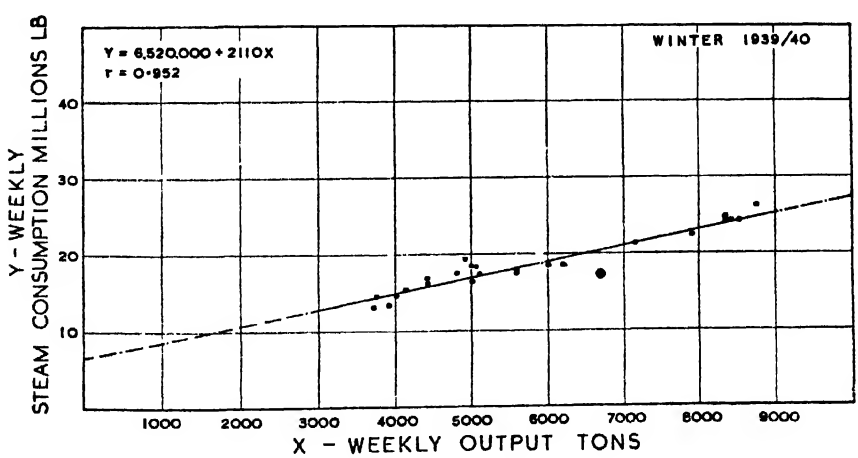


FIG. 386.

Fig. 386 shows the scatter diagram and regression line for the corresponding period three years later, when most of the planned economies had been put into effect. The marginal steam consumption has come down to 2,110 lb./ton—a very remarkable reduction—but the fixed steam has gone up to 6,520,000 lb./

week. This slight increase was thought to be due for the most part to A.R.P. requirements, largely in the form of extra space heating, particularly at week-ends.

Fig. 387 shows the results for the winter period, one year later. War-time fuel economy measures are showing themselves. Almost all the major process or marginal steam economies had been done by 1940, but in 1940/41 a big saving in fixed steam was secured by doing practically all the space and air heating by means of waste heat.

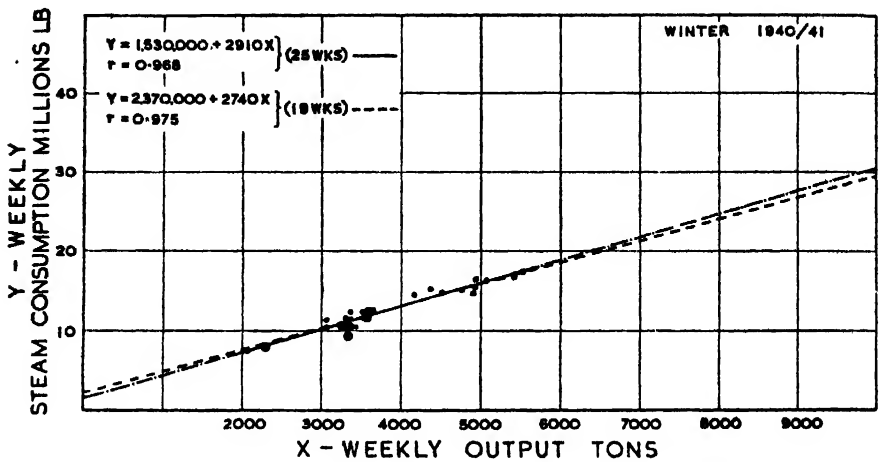


FIG. 387.

The saving in fixed heat is clearly seen in Fig. 387. In fact the full regression line which refers to all 25 points shows an almost incredibly low fixed heat. On the other hand the marginal heat has risen from 2,110 lb./ton in 1940 to 2,910 lb. in 1941. This was thought to be due to difficulties caused by the disorganisation due to the Blitz.

There is here a very interesting point. In Fig. 386 there is one point ringed that lies far outside the grouping of the others. There must surely be something wrong with this point, and should it not therefore be excluded ? But the factory records show nothing abnormal in that week, and if we start excluding weeks because we don't like the look of them we are simply backing our fancy and might confine ourselves to straight guess-work. So this point must stay. But in Fig. 387 there are six points that are ringed. These points look quite good ; they seem to lie nicely about the regression line. But in each of these weeks bombs fell inside the factory. There is therefore every justification for excluding them from the investigation in spite of their good looks.

If these six points are excluded we get the broken regression line which has a fixed steam consumption of 2,370,000 lb./week instead of 1,530,000 lb./ week and a marginal steam consumption of 2,740 lb./ton instead of 2,910 lb./ton. We are certainly right to exclude the bomb weeks and the figures are much more reasonable. Both the regressions are highly significant but the correlation coefficient for 19 weeks is only a little higher than for the 25 weeks and its significance is slightly less.

Why should the bomb weeks show rather lower figures on the whole than the normal weeks ? It would be reasonable to expect the reverse. The probability is that the staff were too busy clearing up the mess to bother about proper meter readings and the punctual changing of charts, so that there was probably some wishful metering.

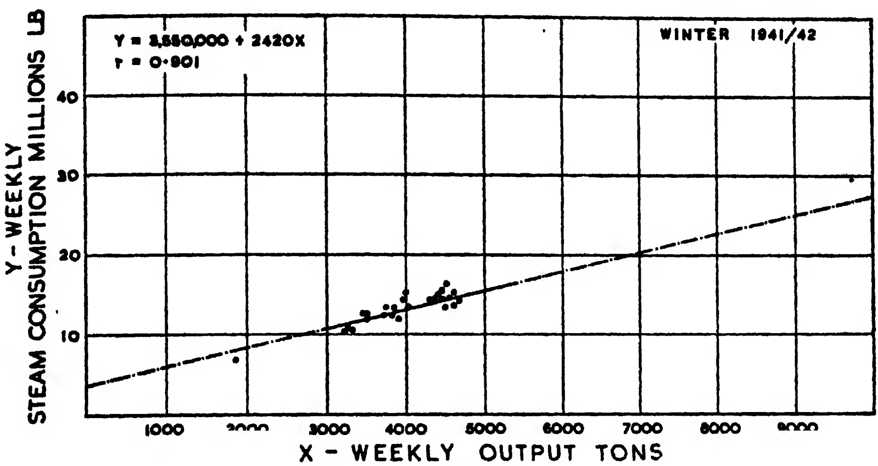


FIG. 388.

Fig. 388 shows the next winter period with stable war-time conditions. Although the correlation is quite good it is not quite so high as previously. This is because the points are more bunched. The output variations were rare and relatively small. But the regression probably gives a fair enough picture of what was happening.

We have seen that while it is very interesting to get from the regression equation the relation between steam consumption and output it is even more interesting to find out the proportions of fixed and marginal steam. The more bunched the points the less reliable must the regression equation be. Now the value of the fixed and marginal steam can only be found from the regression equation. No accountant or technologist can give us this figure, it is the monopoly of the statistician. In order to get a reliable regression equation it is clearly desirable to have as much variation in output as is reasonably possible, and as many points on the scatter diagram as possible. For this reason the figures should be taken out weekly rather than monthly. Although weekly figures are less reliable than monthly figures, the weekly output variations may well be smoothed out when taken monthly.. Unless there is a good output variation it is impossible to get a reliable regression equation.

Fig. 389 shows yet another war-time winter when, due to Ministry of Food requirements, the output was very low and very constant. The scatter is a compact little cluster with a very indefinite trend. The regression line is drawn in, but it is clearly of little use. The correlation coefficient of ·486 confirms this and reference to Table LXXI tells us that this correlation does not satisfy the 100 to 1 criterion.

It is obvious that a small change in the slope of the regression line will have a very large effect on the point where it cuts zero output. This means that the estimate of the fixed steam is not reliable unless the correlation and significance are very high. Even so the fixed component does not give the real steam consumption at zero output unless there are many points at very low outputs.

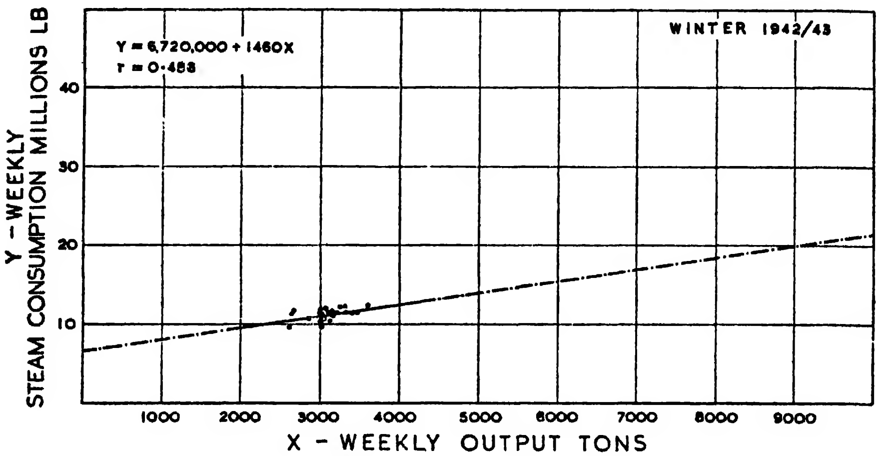


FIG. 389.

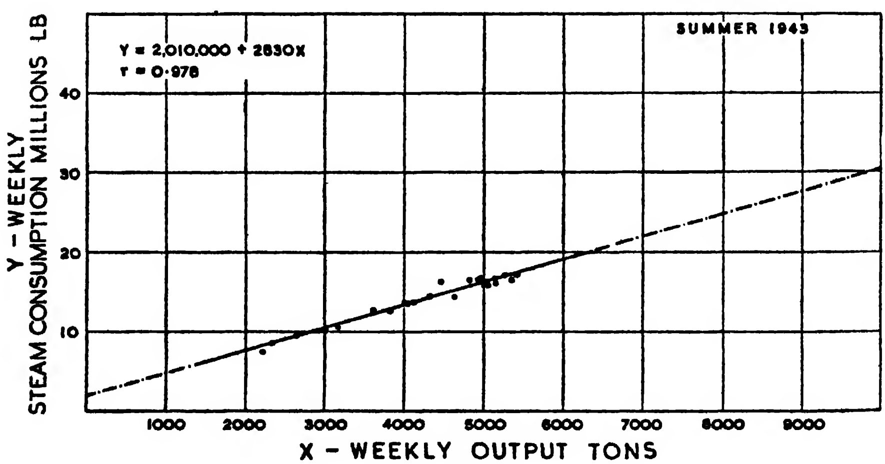


FIG. 390.

During the summer of 1943 the output varied considerably. Fig. 390 shows the very remarkable result. Any eye can see that there is a decided straight line relation. The correlation, both by eye and by calculation, is very high indeed. We can put much more trust in this result. The reliability of the regression line will be dealt with further in the next Section; for the moment we will take Fig. 390 at its face value and compare it with Fig. 386.

It shows that since 1940 the war-time economies, chiefly the recovery of waste heat, have pulled the fixed steam down from some 6,500,000 lb. to some 2,000,000 lb., but that the marginal steam has risen from some 2,100 lb./ton to about 2,800 lb./ton. The increase in marginal steam can probably be quite satisfactorily accounted for by war-time demoralisation, insufficient maintenance, the call-up of skilled men, girls who forget which way to turn a valve, overworked and inadequate managerial staff, etc.

It might be thought that the improvement shown in Fig. 390 was largely due to its being the record of a summer period and that the saving in fixed steam was due to less space heating. But in this factory this is not the case. Almost all the space heating is done by waste heat and there is no significant difference between the steam consumption in summer and winter.

685. TRUSTWORTHINESS OF REGRESSION. Correlation and its significance only tells us the degree of confidence with which we can assert that there is or is not a true relation between steam consumption and output. It does not tell us how accurate the equation is likely to be if we want to use it for prophecy. We have already seen that if the points in the scatter are far away from the origin we can put very little faith in the value of the fixed component, as a true measure of the steam consumption at zero output.

We can find out the degree of trustworthiness, or the "Fiducial Limits" as they are called, of the regression equation by means of formulae which are given in the next two Sections.

686. FIDUCIAL LIMITS OF REGRESSION LINE. The regression equation gives us an estimate of the steam consumption for any output. We wish to find out the degree of trust we can put on the value of the position and slope of this line. This we find by calculating the fiducial limits for the mean steam consumption for any given mean output. We also wish to know the trustworthiness of the estimated steam consumption for any one particular week. We will deal with the first requirement first, namely the trustworthiness of the mean values.

If we calculate the fiducial limits of the mean steam consumption estimate $\bar{Y}'$ we can plot these values on the scatter diagram and we can say that the regression line will lie within the limit curves.

The fiducial limits of $\bar{Y}'$ are

$$\bar{Y}' \pm t\sqrt{V(e)\ \left(\frac{1}{N} + \frac{(X - \bar{X})^2}{SX^2}\right)}$$

The "error variance" term

$$V(e) = \frac{Sy^2 - bSyx}{N - 2}$$

t is the "error ratio".

Here are the figures for the summer of 1944. We first work out the regression equation and find the correlation coefficient as in Section 681.

X	Y
3,235	1,111
3,858	1,331
4,615	1,575
4,752	1,586
5,092	1,561
4,863	1,516
4,464	1,503
4,079	1,230
3,761	1,288
4,838	1,440
4,022	1,321
3,244	1,167
3,165	1,166
3,488	1,139
3,305	1,095
3,463	1,158
3,984	1,392
3,333	1,162
3,051	1,100
3,942	1,293
3,667	1,306
3,716	1,263
3,659	1,316
3,514	1,276
3,515	1,256
3,353	1,227
N = 26: 99,978 = SX	33,778 = SY
3,845·3077 = $\overline{\text{X}}$	1,299·1538 = $\overline{\text{Y}}$

$$
\begin{aligned}
\text{SYX} &= 181{,}966{,}180 \\
\text{SYSX} &= 3{,}377{,}056{,}884 \\
\text{SYSX/N} &= 129{,}886{,}803 \\
\therefore\ \text{S}yx &= 2{,}079{,}377 \\
\text{SX}^2 &= 393{,}141{,}002 \\
(\text{SX})^2 &= 9{,}995{,}600{,}484 \\
(\text{SX})^2/\text{N} &= 384{,}446{,}172 \\
\therefore\ \text{S}x^2 &= 8{,}694{,}830 \\
b &= \cdot 23915096 \\
b\overline{\text{X}} &= 919\cdot 6090 \\
\therefore\ a &= 379\cdot 5448 \\
\text{Y}' &= 3{,}795{,}448 + 2{,}392\text{X}
\end{aligned}
$$

$$
\begin{aligned}
(\text{S}yx)^2 &= 4{,}323{,}808{,}708{,}129 \\
(\text{SY})^2 &= 1{,}140{,}953{,}284 \\
\text{SY}^2 &= 44{,}458{,}788 \\
(\text{SY})^2/\text{N} &= 43{,}882{,}819 \\
\therefore\ \text{S}y^2 &= 575{,}969 \\
\text{S}y^2\,\text{S}x^2 &= 5{,}007{,}952{,}540{,}270 \\
r^2 &= \cdot 863388515 \\
r &= \cdot 9292
\end{aligned}
$$

We can now tabulate the fiducial limits. The computation is quite straightforward. We will re-state the formula for finding the limits between which the regression line should lie. This entails taking the fiducial limits of a number of mean values.

$$\text{Limits of } \overline{\text{Y}}' = \overline{\text{Y}}' \pm t\sqrt{\text{V}(e)\ \left(\frac{1}{\text{N}} + \frac{(\text{X} - \overline{\text{X}})^2}{\text{S}x^2}\right)}$$

$$\text{V}(e) = \frac{\text{S}y^2 - b\text{S}yx}{\text{N} - 2}$$

All these terms have already been individually computed except for t.

$$bSyx = 497{,}285$$
$$Sy^2 - bSyx = 78{,}684$$
$$V(e) = 3{,}278{\cdot}5$$

We therefore have $\pm\, t\sqrt{3{,}278{\cdot}5\left(\frac{1}{N} + \frac{(X-\bar{X})^2}{8{,}694{,}830}\right)}$

$$= \pm\, t\sqrt{126{\cdot}096 + {\cdot}000377063\,(X-\bar{X})^2} = t\sqrt{\theta}$$

$$X - \bar{X} = X - 3{,}845{\cdot}3077$$
$$(X-\bar{X})^2 = X^2 - 7{,}690{\cdot}6154X + 14{,}786{,}391$$
$$\therefore \theta = 126{\cdot}096 + {\cdot}000377063X^2 - 2{\cdot}8998X + 5{,}575{\cdot}4$$
$$= {\cdot}000377063X^2 - 2{\cdot}8998X + 5{,}701{\cdot}5$$

The function t is the "error ratio" and its values for two different sets of odds are given in Table LXXII.

TABLE LXXII. VALUES OF ERROR RATIO "t"

N	20 TO 1	100 TO 1
100	1·985	2·53
52	2·005	2·68
50	2·01	2·685
40	2·025	2·71
30	2·05	2·74
27	2·06	2·78
26	2·06	2·79
25	2·07	2·80
20	2·10	2·87
15	2·15	3·00
10	2·30	3·35

We see from Table LXXII that the error ratio t is 2·06 at 20 to 1. This means that the fiducial limits that we are about to find are such that 95 per cent. of the estimates will lie within the limits.

We can now tabulate for various values of X thus. (As all the Y's are in units of 10,000, the limits, $t\sqrt{\theta}$, computed in this tabulation must be multiplied by 10,000 before being added to or deducted from $\bar{Y}'$.)

X	$\cdot000377063X^2$	$- 2\cdot8998X$	θ	$\sqrt{\theta}$	$t\sqrt{\theta}$	$\bar{Y}'$	Limits of $\bar{Y}'$	
0	0	0	5,701·5	75·51	155·55	3,795,500	2,240,000 to	5,351,000
1,000	377·1	− 2,899·8	3,178·8	56·38	116·14	6,187,500	5,026,100	7,348,900
2,000	1,508·3	− 5,799·6	1,410·2	37·55	77·35	8,579,500	7,806,000	9,353,000
3,000	3,393·6	− 8,699·4	395·7	19·89	40·97	10,971,500	10,561,800	11,381,200
4,000	6,033·0	− 11,599·2	135·3	11·63	23·96	13,363,500	13,123,900	13,603,100
5,000	9,426·6	− 14,499,0	629·1	25·08	51·66	15,755,500	15,238,900	16,272,100
6,000	13,574·3	− 17,398·8	1,877·0	43·32	89·24	18,147,500	17,255,100	19,039,900
7,000	18,476·1	− 20,298·6	3,879·0	62·28	128·30	20,539,500	19,256,500	21,822,500
8,000	24,132·0	− 23,198·4	6,635·1	81·46	167·81	22,931,500	21,253,400	24,609,600
9,000	30,542·1	− 26,098·2	10,145·4	100·72	207·48	25,323,500	23,248,700	27,398,300
10,000	37,706·3	− 28,998·0	14,409·8	120·04	247·28	27,715,500	25,242,700	30,188,300

These limits are plotted as the inner broken lines in Fig. 391.

This means that if we had 100 samples of 26 weeks (under the same conditions) instead of only one set of 26 weeks we should expect the regression lines of 95 of them to lie within this band.

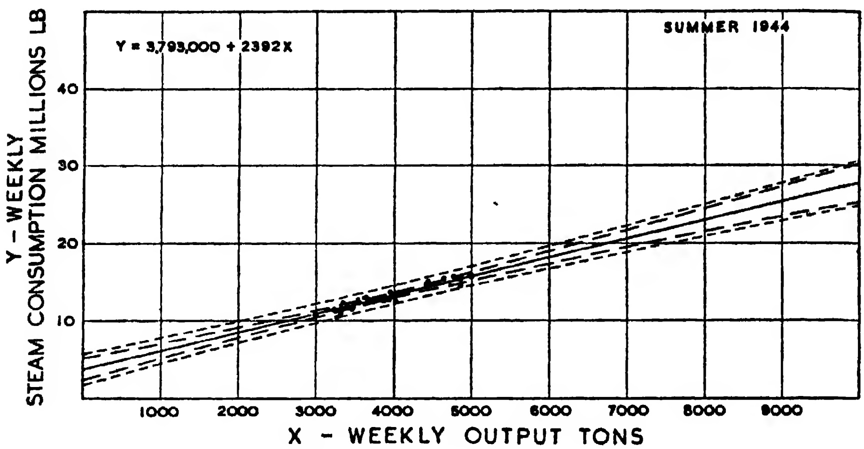


FIG. 391. FIDUCIAL LIMIT BANDS

687. FIDUCIAL LIMITS OF INDIVIDUAL WEEKS. Any individual week must of course be given wider limits, as can be seen by an examination of any of the scatter diagrams that have been given.

The fiducial limits for a single estimate, that is to say the limits within which the estimate of the steam consumption for any one week will lie, are obtained from the following expression :—

$$Y' \pm t\sqrt{V(e)\ (1 + \frac{1}{N} + \frac{(X-\bar{X})^2}{Sx^2})}$$

It will be seen that this simply adds $V(e)$ to the quantity under the square root sign. Now $V(e) = 3{,}278{\cdot}5$, so we can retabulate by adding 3,278·5 to θ and calling the sum ϕ.

X	θ	ϕ	$\sqrt{\phi}$	$t\sqrt{\phi}$	Y′	Limits of Y′	
0	5,701·5	8,980·0	94·76	195·21	3,795,500	1,843,400 to	5,747,600
1,000	3,178·8	6,457·3	80·36	165·54	6,187,500	4,532,100	7,842,900
2,000	1,410·2	4,688·7	68·47	141·05	8,579,500	7,169,000	9,990,000
3,000	395·7	3,674·2	60·62	124·88	10,971,500	9,722,700	12,220,300
4,000	135·3	3,413·8	58·43	120·37	13,363,500	12,159,800	14,567,200
5,000	629·1	3,907·6	62·51	128·77	15,755,500	14,467,800	17,043,200
6,000	1,877·0	5,155·5	71·80	147·91	18,147,500	16,668,400	19,626,600
7,000	3,879·0	7,157·5	84·60	174·28	20,539,500	18,796,700	22,282,300
8,000	6,635·1	9,913·6	99·57	205·11	22,931,500	20,880,400	24,982,600
9,000	10,145·4	13,423·9	115·86	238·67	25,323,500	22,936,800	27,710,200
10,000	14,409·8	17,688·3	133·00	273·98	27,715,500	24,975,700	30,455,300

These limits are shown as the outer dotted lines in Fig. 391.

In Fig. 392 are the regression lines for the half years from 1940 to 1944, omitting only that for the winter of 42/43 where the correlation was unsatisfactory. The inner fiducial limit curves from Fig. 391 have been drawn in and it will be seen how nicely the regression lines lie nearly within the 20 to 1 area.

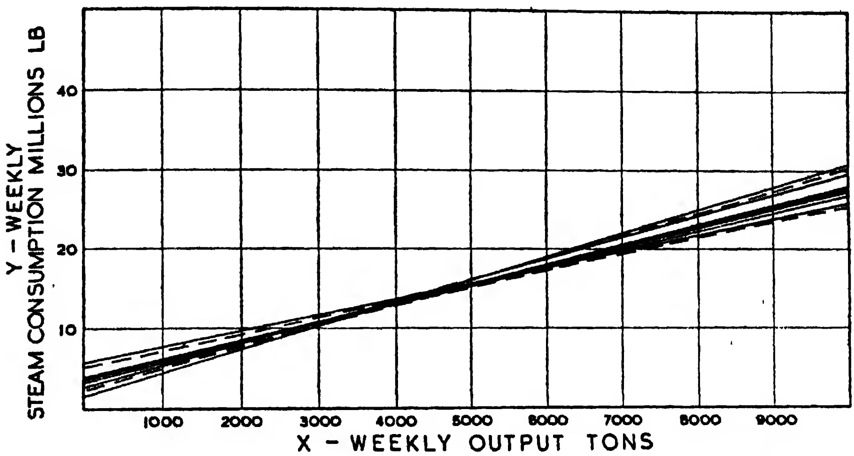


FIG. 392. SEVEN REGRESSION LINES AND INNER FIDUCIAL BAND FROM FIG. 391

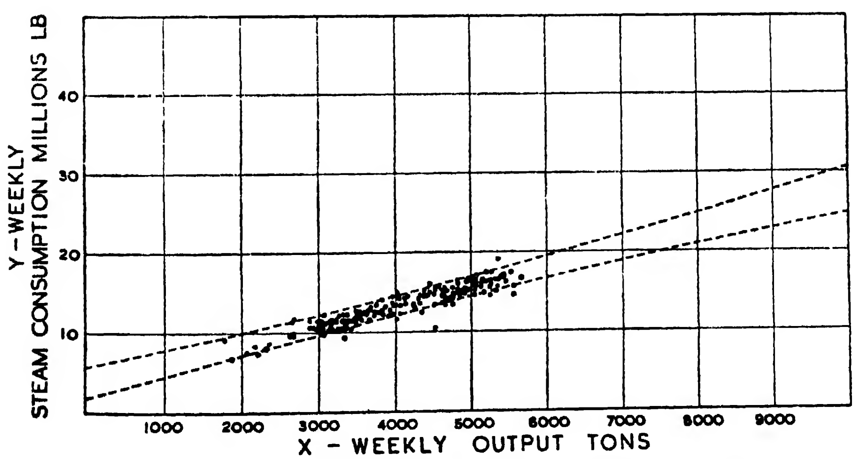


FIG. 393. 208 WEEKLY POINTS AND OUTER FIDUCIAL BAND FROM FIG. 391

In Fig. 393 are all the points for eight consecutive half years, some 208 in number, and the outer fiducial limit curves are drawn in from Fig. 391. These outer fiducial lines are such that the odds are 20 to 1 on the estimated steam consumption for any individual week lying between them. If the conditions had remained exactly constant over the whole period we should therefore expect to find that about 11 points would lie outside the fiducial band. Actually

15 points lie outside. So it ties up pretty well considering that conditions have have not remained constant. Small improvements in steam use have been continuously made, but there has been a steady and continuous deterioration due to war difficulties. In view of this, theory and practice agree remarkably well.

688. THE PRACTICAL VALUE OF REGRESSION. The regression equation does not give us an exact figure for fixed steam and marginal steam. If the correlation is significant it gives us a very good estimate of the marginal steam and a less reliable estimate of the fixed steam. In either case it gives us nothing more than a good bet, but in spite of its limitations it is a very useful tool because it does give us information that can be obtained in no other way. The accountant cannot tell us the cost of our fixed steam at all, but he can go into the witness box and swear to the total cost of steam. The statistician can give us an estimate of something that the accountant cannot approach, but the statistician can only go into the bookie's office and come out smiling.

One important lesson is clear from this brief investigation of regression, namely the extreme caution that should be used when interpreting the results of experiments.

689. THE REGRESSION LINE AND THE WILLANS LINE. If steam consumption figures for an engine or turbine are plotted against output they will be found to lie on a straight line almost without scatter, provided the measurements have been made with accuracy. The relation between steam consumption and output can be calculated as a regression equation and the correlation will be very significant. There is no doubt that there is a relation. There is a fixed heat consumption which makes good the losses and this lost heat cannot be calculated. It can be measured, or it can be estimated by an experienced engineer. If however a few widely spaced points are taken so that the Willans Line can be drawn, or better still calculated as a regression line, the fixed lost steam can be found quite accurately.

Now it is not only an engine that has a Willans line. Every piece of steam-using plant has one, or its equivalent. Every vat, or pan or evaporator has a fixed steam to make good the losses and a marginal steam that does the work. The regression line of a factory takes account of the Willans lines of all the engines, vats, kiers, pans, coppers, etc., in the factory, and the interaction of all these goes to impart a large element of chance or error into the combined data from which we calculate the factory regression equation. A single Willans line is a certainty. A multitude of random Willans lines is only a good bet.

* * *

All the foregoing discussion has dealt with steam consumption related to output. The regression analysis could have been on costs or wages or maintenance or waste of materials. The appropriate regression equation can usefully be ascertained for every operation or cost that might comprise a fixed but dimensionally unknown component and a marginal component in its make up. By making a regression analysis the existence of a totally unexpected fixed or marginal component is sometimes discovered which may put a completely different complexion on a problem.

* * *

CHAPTER 22

SAVING POWER AND ELECTRICITY

To give light to them that sit in darkness.
LUKE. I, 79. A.D. 60

AS far as the author knows there is no book which deals with the saving of power, no book or curriculum which deals with the saving of electricity. As steam is responsible for the production of the bulk of the power and electricity in this country, economy of power and electricity is economy of steam.

690. THE INEFFICIENCY OF POWER. The power saving field is enormous. The author thinks that he is greatly understating the case by saying that nine-tenths of the coal burnt for power generation in this country is entirely wasted. For example, take a textile mill electrically driven from the grid :—

	Per cent.	Per cent.
The average efficiency of textile machinery seldom exceeds		20
The efficiency of the motors is probably about	85	
∴ The efficiency at the motor terminals is 20 × ·85		= 17
The distribution loss in 1957 was about ..	12	
∴ The efficiency at the power station switchboard was 17 × ·88		= 15
The average efficiency of power generation was about	25	
∴ The overall efficiency of power generation and use in the mill was 15 × ·25		= 3·75

So that a very small saving of power energy results in a huge saving of steam and coal energy. No further excuse is needed for dealing with the subject in a book devoted to the efficient use of steam.

691. THE POWER LOAD. In Section 123, Chapter 3, it was pointed out that modern plants must always tend to increase their power load and to decrease their steam demand. If the factory buys its power all it has to worry about is coming to a good arrangement with the power company regarding rates, maximum demand and power factor. It is then an easy problem to see how much it is worth spending to save electricity.

If the factory generates its own power, whether by direct steam engine or by electrical generation, it is very important to keep the load within bounds lest the power plant be unable to carry the load and calls for costly renewal or supplement.

If the factory can barely use, for legitimate purposes, all the exhaust steam from its power plant, it is prevented from making possible great steam savings unless the power load can be cut simultaneously.

From every point of view therefore it is of the greatest importance that the power load be cut to the lowest possible point.

692. POWER FACTOR. Of all the subjects regarding power saving on which there is misunderstanding the question of power factor is probably first. Many instances have been met where factory managements have thought that by improving power factor by ·1 they have secured a power saving of 10 per cent. Much of this misunderstanding is due to the stressing by the electricity supply companies of the importance of power factor and of their habit of charging for kVA.

It is extremely difficult to explain power factor in simple non-mathematical language. The following is an attempt which does not aspire to explain it, but to give an inkling only of what power factor means and the way in which it affects electrical plant. The author is not an electrical engineer and is consequently prepared to simplify in a way that might make the electrically trained expert shudder.

693. KILOWATTS AND KILOVOLTAMPS. The pressure or potential of electricity is measured in units called " volts ". The rate of flow of electricity is measured in units called " amperes ". (These names are tributes to the memory of two great scientists—Volta, an Italian who, in 1800, made the first electric chemical cell or battery—Ampère, a Frenchman who, in 1820, first worked out the mathematics of electro-magnetism.) The " power " in an electric circuit is the product of the " flow " and the " pressure ". That is to say, a given flow, or number of amperes, of electricity flowing at a certain pressure or number of volts will give double the power if the voltage is doubled, or double the power if the voltage is kept constant and the flow or amperes be doubled.

Electrical power is found by multiplying the volts by the amperes. The product V × A = VA is in power units which are called " watts ", a compliment to the memory of James Watt who, though he did no work on electricity, was the true father of power. So that circuits carrying the following varieties of current are all carrying the same power :—

$$\left.\begin{array}{rcrl} 1 \text{ volt} & \times & 1{,}000 & \text{amperes} \\ 10 \ \text{,,} & \times & 100 & \text{,,} \\ 100 \ \text{,,} & \times & 10 & \text{,,} \\ 1{,}000 \ \text{,,} & \times & 1 & \text{,,} \end{array}\right\} \begin{array}{l} = 1{,}000 \text{ VA} = 1{,}000 \text{ W.} \\ = 1 \text{ kVA} = 1 \text{ kW.} \end{array}$$

Now this simple relation between voltamps and watts, or kilovoltamps and kilowatts as they are usually called because the figures would otherwise be inconveniently large, only holds good with direct current (D.C.) electricity.

694. THE WATER ANALOGY. Direct current electricity is exactly the same in principle as water flowing through a pipe. If we pump water into a pipe by means of a reciprocating pump provided with an inlet and exhaust valve we can drive a water engine at the other end of the pipe, provided the engine is fitted with a valve gear. The water exhausted from the engine can be returned to the suction of the pump. Such a method of transmitting power is completely analogous to direct current electrical power transmission. The cylinders of the pump and engine are equivalent to the poles on the dynamo and motor, and the valve gears are exactly reproduced by the commutators. The water motor or engine can run at any speed, quite independent of the speed of the driving pump,

provided there is a return for the surplus water. The work done by the engine will be the flow of water to the engine multiplied by its pressure (apart from losses due to mechanical imperfections).

Now suppose we have a three cylinder pump each of whose cylinders is connected by a separate pipe to the three cylinders of a water engine—Fig. 394. If the cranks in both machines are at 120° neither pump nor engine will need any valves. The water in each pipe flows to and fro—it alternates. Provided the engine is not so overloaded as to stall, the engine will keep exactly in step with the pump. Such an arrangement is exactly equivalent to a three phase alternating current generator driving a three phase synchronous motor. The cylinders are equivalent to the poles, there are no valves and no commutators.

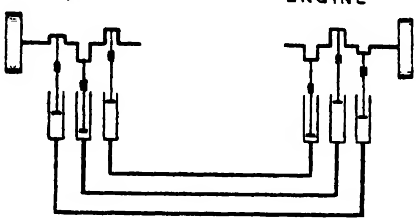


FIG. 394. THREE PHASE ELECTRICAL WATER ANALOGY

The work done by the engine will be the quantity of water flowing to and fro in each pipe multiplied by its pressure and by the number of pipes or phases. This is the same as saying that the power is volts × amps × 3.

From now on the analogy becomes less precise but should still be helpful.

Suppose we allow a small amount of air to get entangled, foam-like, in the pipes, and that we assume that pump and engine are far apart. When the pistons of the pump descend they will displace the water but, owing to the compressibility of the air now mixed with the water in the pipe, the pressure impulse reaches the engine slightly delayed—it lags behind the initial pressure—due to the cushioning effect of the air. If the speed is such, in comparison to the pipe length, that the pressure wave is in tune with pipe length, there will be a point when the pressure wave never reaches the engine because, before the pressure wave has reached the engine, the pump piston will have started rising, and the pressure wave will start changing direction. When this happens the engine stops work. Although the descending piston did much work on the water, this energy was stored in the compression of the air bubbles and is returned to the piston as it rises on the upward stroke. But this energy conversion and reconversion is not perfect and gradually the pump and pipes will heat up.

When a voltage wave from an alternating current generator flows into a motor the ampere or current wave lags behind the voltage wave just like the water wave in the aerated pipe. If the conditions are suitable the current wave can lag so far behind the voltage wave that the current has a maximum value when the voltage is zero. Current of this kind does not produce power. If an " A " current arrives when a " V " voltage is 0 then the simultaneous

$V \times A = 0$, so that there are no watts. Such a lagging current is therefore called "wattless" current. The measure of the amount by which the current lags behind the voltage is called the "power factor". When the power factor is 1·0 the volts and amperes flow simultaneously. When the current lags so far behind the pressure that the amperes are a maximum when the volts are zero the power factor is 0 and such a current will not generate any power, but, like the water in the pipe, the passage of this wattless current will heat all the wires through which it flows.

695. OHM'S LAW. The flow of electricity through a circuit depends on the voltage of the supply and the resistance of the circuit. The resistance depends on the size of the wires, the metal of which they are made and their temperature. The law which the flow of electricity obeys was discovered by a German professor called Ohm in 1827. His discovery was so coldly received by the scientific men of the world that Ohm resigned his professorship in a huff. (Ohm's law was actually discovered by Cavendish in 1781, but Cavendish was a queer chap and did not publish his discovery.) Ohm's law states that the flow of current (amperes) will be directly proportional to the pressure (volts) and inversely proportional to the resistance (ohms). This means that if the volts are doubled with the same resistance the current flow will be doubled. If the voltage is kept constant and the resistance is doubled the current flow will be halved.

$$\text{Current in amperes} = \frac{\text{Pressure in volts}}{\text{Resistance in ohms}}$$

$$A = \frac{V}{O}. \qquad O = \frac{V}{A}. \qquad V = A \times O.$$

Ohm's law only applies when there are no disturbing factors. When current flows into a D.C. motor there is a very large disturbing factor. The motor by its operation acts as a generator and produces a pressure that opposes the input pressure. This must be allowed for. This output pressure generated or induced by a motor in opposition to the input pressure is called the Back E.M.F. (or back electro-motive force). Suppose a motor is supplied with current at a pressure of 100 volts. At starting, the motor is momentarily at rest, no work is being done, no opposing pressure is being generated, and Ohm's law applies. We will say that the current momentarily is 100 amperes. This means that the resistance of the motor is 1 ohm. Now when the motor is running at full speed on full load it may be taking 10 amperes. The voltage has not changed, the resistance has not changed, so that the reduction of current flowing must be due to the voltage that the motor is generating in opposition to the input supply. So that of the input 100 volts, 90 must be being used to oppose and overcome the opposing voltage leaving only 10 to force the current through the motor. As the resistance of the motor is 1 ohm, 10 volts will result in the flow of 10 amperes.

696. THE HEATING EFFECT OF ELECTRICITY. Now in the motor we have been considering the input current is 10 amperes at 100 volts or a current with a power of 1,000 watts. We have seen that 10 volts are used up in forcing the current through the resisting wires, so that $10 \times 10 = 100$ watts are not

being used for power production but for forcing the current through the machine. This current must appear somehow, and as it does not get converted into mechanical work it must appear as heat.

This 10 volts × 10 amperes which will simply cause heating of the machine is the heat loss in watts. Before we can find the value of this heat loss we must know the voltage that is being used to overcome the resistance. This resistance-overcoming voltage is not always easy to find directly but Ohm's law above, in its third form, gives us a way round. $V = A \times O$. So that we can say truly the power dissipated in heat in a circuit is $A \times V = A \times A \times O = A^2O$. In ordinary parlance among electrical engineers the letters I and R are used instead of A and O for current in amperes and resistance in ohms, so that the heat loss in an electric circuit is equal to I^2R.

In an electric radiator all the electric energy that we put into it is converted into heat. When the radiator is switched on it does not rush round the room or whizz like a catherine wheel. So that its I^2R is equal to its $AV = W$. If however we put current into an electric motor most of the current is turned into mechanical work and only a small part goes in heat. Let us consider our notional motor again. At the moment of switching on, the motor is stationary for an instant, there is a big rush of current and momentarily none of it is turned into work, it must all therefore be turned into heat.

We have 100 volts × 100 amperes = 10,000 watts.

We also have a heat loss equal to $I^2R = 100 \times 100 \times 1 = 10{,}000$

When the motor is up to speed on full load we have an input of

$$100 \text{ volts} \times 10 \text{ amperes} = 1{,}000 \text{ watts}$$

but the heat loss will only be

$$10 \times 10 \times 1 = 100 \text{ watts}$$

leaving 900 watts converted into work.

697. WATTLESS CURRENT. The conditions that we have been considering apply to direct current machines. Suppose we consider our motor now to be an alternating current machine and that the amperes are lagging so far behind the volts that the power factor is ·8. In order to give the full load the motor will now call for $\frac{10}{\cdot 8} = 12\cdot 5$ amperes.

900 watts will still be converted into work and the amount of energy lost in heat will be

$$12\cdot 5 \times 12\cdot 5 \times 1 = 156$$

So that at a power factor of ·8 our motor will be converting

900 watts into power
and 156 „ „ heat

1,056

The effective power taken by the motor will be

$$V \times A \times \text{power factor} = 100 \times 12\cdot 5 \times \cdot 8 = 1{,}000 \text{ W} = 1 \text{ kW}$$

The volts × amperes taken by the motor as measured by voltmeter and ammeter will be

$$V \times A = 100 \times 12 \cdot 5 = 1{,}250 \text{ VA} = 1 \cdot 25 \text{ kVA}$$

250 of this VA is wattless and causes an extra heat loss of 56.

The supply generator has to supply 1,000 effective watts and 56 heating wattless units, a total of 1,056, but the current flowing as measured by voltmeter and ammeter is 1,250 volt-amps. The gain therefore by improving the power factor from ·8 to 1·0 will not be 25 per cent. or even 20 per cent., but 5·3 per cent. (These figures are purely notional to suit the crude example given.)

The load that cables and generator can carry is limited only by the amount of heating they can stand. We see that if the power factor in the motor circuit is ·8 instead of 1·0 the heating current will be increased (in this notional example) from 100 to 156. This is the reason why electricity supply companies are so concerned about power factor.

698. CAUSE OF LOW POWER FACTOR. Low power factor is the fault of the load, not the fault of the generator.

In an induction motor the magnetising current is taken from the line, and as, of itself, it produces no power, it has a zero power factor. (This statement must be taken on trust and may not be approved by the pure electrician, but it is near enough the truth for the present argument.)

When the motor is on full load the magnetising current represents a small fraction of the total current so that the power factor will be high. As the load on the motor drops, the magnetising current represents an increasing proportion of the total current so that the power factor will drop. At no load the power factor will only be ·1 or ·2 and were the motor to be perfect and to have no mechanical losses the power factor at no load would be zero.

In a synchronous motor the magnetising current is supplied from an external source, usually from an exciter on the motor shaft. If the exciting current is just suited to the load the power factor will be unity. If the exciting current is too small, the magnetic system will *borrow* some excitement by induction from the stator alternating current. This borrowed current will be just the same as the exciting current of an induction motor and will have a zero power factor. The proportion of borrowed exciting current will lower the power factor of the motor circuit to an extent proportional to its fraction of the total current. If, on the other hand, the synchronous motor is over-excited from its own exciter a current will be *lent* to the stator current. Whereas the borrowed current lagged behind the main current, the lent current will lead in front of the main current, so that the synchronous motor can operate with a leading power factor. Such a leading power factor can be used to counterbalance a lagging power factor in other parts of the local network and will improve the power factor of the whole system.

Were all the load on an electrical network to consist of over-excited synchronous motors the whole system would have a leading power factor which would be just as bad as an equivalent lagging power factor. So that power

factor improvers—synchronous motors or condensers—should not be used wholesale but only just sufficiently to effect the improvement required. Power factor improvement is expensive.

699. THE PRACTICAL EFFECTS OF POWER FACTOR. If current is being charged for on a kVA basis it clearly pays greatly to improve the power factor, because when receiving 1,000 useful watts and 56 useless units we are paying for 1,250 volt-amps.

If current is being bought on a kW basis there is very little to be gained.

Where current is being generated in the factory, power factor improvement only pays good dividends if motors, cables, etc., are so over-heating as to call for duplication or increase in size. But of course it is always beneficial to improve the power factor. All the electrical plant runs cooler, will carry a bigger load and, to the small extent of the extra heating saved, will save steam.

There is one case where power factor improvement may save a lot of steam, and this case fairly often crops up ; where the load cannot quite be carried by one generating set due to heating of the generator. In Fig. 39 the steam consumptions of two 175 kW sets were shown and we see that by simply starting the second set an immediate increase in steam consumption of 3,000 lb./hr. takes place before one single extra unit of electricity is generated.

If one generator can produce 175 kW with a power factor of ·9 with just the permissible rise of temperature, it would only be possible to generate 156 kW were the power factor to drop to ·8. If the load must be 175 kW we get—

Load kW	*Power Factor*	*Engines Running*	*Steam Consumption lb./hr.*
175	·9	1	7,000
175 { 156	·8	2	{ 6,600
19 }			3,500
			10,100

We can therefore say quite categorically that in most cases of factory power generation it only pays to spend a lot of money on power factor improvement if it prevents the running of an additional generating set. But it is very good policy to improve power factor by all possible cheap means ; one of the easiest means is to see that motors are not very lightly loaded or to run those that are under-loaded under different conditions.

Recapitulating : The voltmeter and ammeter record volts and amperes. The amperes cause heating of the cables and machines. The ammeter does not tell us whether the amperes are wattless or are effective. The power factor meter tells us what proportion of the amperes are powerful. So that the amperes tell us the load that can produce heating ; volts × amps. × power factor tells us the effective power load of the circuit.

700. STAR AND DELTA. There are two ways in which the windings on a three-phase alternating current motor can be connected to the supply line. These are called " star ", Fig. 395, and " delta ", Fig. 396. When connected in delta the full line voltage acts on each winding, but when connected in star the effective voltage across each coil is the resultant of the voltage in all three. Thus when the voltage in A is a maximum in one direction it is opposed by a small voltage in both B and C in the opposite direction. The result is that the voltage acting on each coil in star is ·58 of the voltage across the line or the voltage in delta.

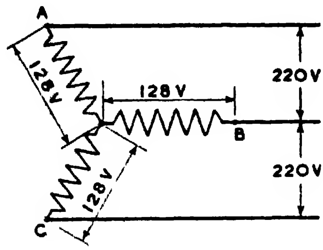


FIG. 395. THREE PHASE STAR CONNECTION

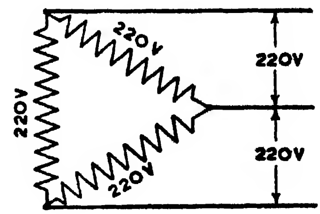


FIG. 396. THREE PHASE DELTA CONNECTION

When a stationary motor is switched on to the line the great initial current rush might be very objectionable, not only to the motor and the switchgear, but because of the surge of current on the line which would throw demands back to the engine governor. This rush of current is greatly minimised by starting the motor on star connection and changing over to delta after a good speed has been reached so that there is plenty of opposition voltage to prevent a rush from the line.

On star the motor will only carry about one-third or so of the load that it will accept on delta, so that motors are almost always built as delta machines thus enabling them to be much smaller and cheaper.

There are many drives that require much power to get them going but take little power to keep them running. Machines with large flywheels like presses or punches, ball mills, rotary kilns, etc., where heavy masses have to be set in motion. The initial movement is done on star to limit current rush, but the motor would not develop enough power on star to bring the heavy machine up to speed. The final acceleration is done on delta. Once they are up to speed such machines often require only one-third or a quarter of their rating to keep them going and might with advantage be switched back to star.

701. ADVANTAGES OF STAR CONNECTION. Star connection, for reasons that need not be gone into, is more efficient and gives a better power factor than delta connection. Fig. 397 shows the efficiency and power factor

of a typical 12 H.P. motor on star and delta connection. We can tabulate the following striking comparison :—

Load H.P.	*Connection*	*Efficiency per cent.*	*Power factor*	*Current amps*
3	Star	85	·79	4·38
	Delta	77	·53	7·19
4	Star	86	·83	5·50
	Delta	80	·60	8·15

We see that there is an advantage to be got from using star where it is possible. But the possibility is relatively rare. It is most effective if the running load is about a quarter or one-third of the rating of the motor. If the load goes up to half, there may be trouble ; the star motor may not take it.

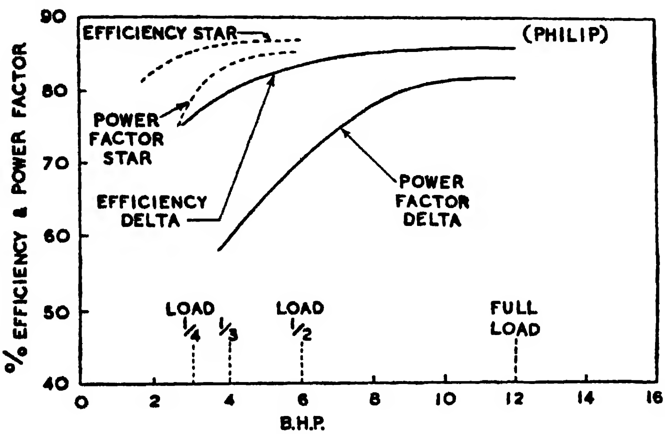


FIG. 397. MOTOR PERFORMANCE CURVES

Where a motor is simply far too big and has not got to start some very heavy machine, it may be possible to start in delta and run on star. But where the starting load is heavy it will be necessary to start in star, accelerate in delta and go back to star, if the load is so light that this is possible.

Star running will almost certainly call for a modification of the starter. The protective devices are generally arranged in the delta position only, and the ammeter, if fitted, will probably only be in circuit on delta. Modifications to the protection relays or trips are usually quite simple to carry out and the saving may be worth the trouble entailed.

702. OVERSIZE MOTORS. The size of motors is important, but there are points that require clearing up. A motor is more efficient on full load than on half load, but a large motor on half load may be more efficient than a small motor on full load. Table LXXIII in the Appendix gives results obtained on motors of one of the leading electrical machinery makes, taken from stock in the author's factory. Each motor in turn was made to drive the same generator,

whose load was raised by increments of 1 H.P. It shows that overmotoring up to about twice too big is probably the ideal for drives below about 15 H.P. The power used by the larger lightly loaded motor is about the same as by the fully loaded smaller motor. In some cases it is less. The power factor is better and, more important, speed is maintained. The small motors drop considerably in speed on full load. The double size motor takes half load with very little speed reduction. The oversize motor will run cool and will be very free from breakdown, but it represents a bigger capital value.

One of the important things visible from Table LXXIII is that a factory can justifiably reduce the number of spare motor sizes kept in stock. Instead of stocking 3, 5, 7½, 12½, 18 and 25 H.P. motors, very little power, if any, would be lost by stocking only 5, 12½ and 25. The extra capital sunk in bigger motors might well be offset by the fewer spares that need be carried.

In Table LXXIII the heavy line shows the boundary between correct load and overload. An examination of the table shows that it is most important not to overload motors. The efficiency and speed drop off badly, the motor gets hot and is prone to break down.

It will be seen that a large motor on star can in some cases beat a motor one-third or one quarter its size on delta.

703. UNIT DRIVE OF LINESHAFT. The modern practice is to drive each machine or piece of plant with its own separate motor. This has many advantages ; it is clean, tidy, convenient and safe. On the face of it, it might seem extravagant because each motor must be big enough to handle its own peaks and starting load. There will therefore be a larger installed horsepower than with belt drive from a line shaft driven by one large motor.

Lineshafts are unsightly, dirty, obstruct the light and cast shadows, are hazardous, require belt maintenance, give rise to belt slip, suffer from belt breakage and call for extra power to drive the shaft and bend the belts round the pulleys. Against these obvious disadvantages appears the seeming advantage of one properly sized motor of a much smaller installed horsepower.

On the other hand the motor on an individually driven machine can be stopped when the machine is stopped, whereas if only one machine on a lineshaft is running, the big motor must run all the time. Table LXXIII shows how wasteful this might be.

From the power point of view there would seem therefore to be little in any argument against individual drive. The choice between lineshaft and individual drive must be made on other considerations, which are probably overwhelmingly in favour of individual motors.

When the proper machine speed is known, direct drive is desirable on many counts. Where speed changes may be desirable from time to time, or a row of like machines must be driven at slightly different speeds, the lineshaft is the obvious choice. Such an example existed for many years in the author's factory where a row of 30 identical machines were operated by 30 girls on piecework. Each girl was allowed to have her machine running at the speed she liked best and variations of 20 per cent. were not uncommon. The lineshaft with split pulleys provided the ideal arrangement.

704. EXCESSIVE SPEED. This is the real villain of the piece, and the effect of speed deserves real study. The power taken by most ordinary machines is directly proportional to the speed at which they are run. This type of machine includes machine tools, elevators, mixers, conveyors, reciprocating pumps (at constant pressure), etc.

The other class of machine includes fans, blowers and centrifugal pumps. If these machine have a free discharge, the power taken varies as the cube of the speed. If the discharge is throttled by valve or damper, the power taken varies as the square of the speed. In these classes of machine the scope for power saving may be really great.

705. FANS. In the case of fans, only too often does one find a bit of cardboard or a sack hanging over the outlet to temper the blast. When a fan is installed it is generally done with little real thought except to make sure that it is not too small.

In the author's factory many air-conditioning fans were found to give satisfactory air supply when reduced in speed by 14 per cent. These plants have undamped discharges where the power is proportional to the cube of the speed. As the speeds were reduced in the ratio of 7 to 6 the power saving was

$$\begin{array}{r} 7 \times 7 \times 7 = 343 \\ 6 \times 6 \times 6 = 216 \\ \hline 127 \text{ or } 37 \text{ per cent.} \end{array}$$

When supplying fans contractors are torn between two desires ; first to quote for the cheapest plant to ensure getting the order ; second to make quite sure that their plant has a margin to meet requirements. For example, a boiler maker will calculate the requirements for an induced draught fan. He will then add a margin to protect himself from his customer. The fan maker receives an enquiry from the boiler maker for a fan of a certain duty. The fan maker adds on a margin to protect himself from his customer. The fan maker sends his enquiry to the motor maker who in turn also adds a margin. In this way the power specification snowballs. The fan starts work with far too big a motor, the fan is also too big and the fan has to be run continually against partly closed dampers. To remedy this, the eventual user–customer must insist on proper subcontracting co-operation.

The power taken by a damped fan varies as the square of the speed. If therefore the speed could be reduced, possibly to such a speed that the dampers could be wide open most of the time, and if this speed reduction were again to be 14 per cent. the power saving would be

$$\begin{array}{r} 7 \times 7 = 49 \\ 6 \times 6 = 36 \\ \hline 13 \text{ or } 26{\cdot}5 \text{ per cent.} \end{array}$$

Small windmill type fans almost always run too fast, and present a considerable problem because there is generally no way by which they can be run slower. The blades are usually of indifferent aerodynamic shape and great

eddies and disturbances occur. In the author's factory there are many propeller fans, both large and small, where a handkerchief held in front of the discharge is sucked in towards the propeller boss.

706. CENTRIFUGAL PUMPS. Centrifugal pumps follow the same power-speed laws as fans. If there is open discharge, the power varies as the cube of the speed, if the discharge is throttled the power varies as the square of the speed.

If centrifugal pumps are direct driven there is little that can be done about speed variation. If they are belt or V–rope driven it is easy to slow them down so that they need not be throttled. When the impeller wears so that the outlet pressure drops, it is also easy with such drives to raise the speed a little and thus delay major repair. In the author's factory good results are being obtained from chain driven centrifugal pumps. The drive is extremely efficient, it cannot slip, the sprockets can be quickly changed to give any desired speed but it is very expensive.

Centrifugal pumps are often used in positions where they are quite unsuitable, namely where the output is low and the pressure is high. The smaller the output the lower must be the pressure to get anything like reasonable efficiency. If a small quantity is to be pumped against a high head it either means that the impeller must be of large diameter or the pump must be a multistage small diameter pump. The small multistage centrifugal pump is an abomination, it is quite remarkably inefficient and unsatisfactory. A large diameter single stage pump usually means a large pump, otherwise the impeller width approaches the clearances in capacity, and when this is the condition much of the liquid simply circulates round the clearance, and the efficiency is again very low. If a reasonably dimensioned single stage centrifugal pump is used for small quantities it will always be too big and need throttling. This just means that the impeller will churn, waste power, generate heat and possibly damage the product.

Small output high head centrifugal pumps should be replaced by reciprocating pumps. In the author's factory a high pressure centrifugal pump on small quantities taking 15 H.P. was replaced by a reciprocating pump using 3 H.P.

For many duties, particularly where the lift is high and the quantity is small, centrifugal pumps are quite unsuitable. They are too often selected and installed because they are small and cheap. For small outputs and high pressure the reciprocating pump is supreme although it is looked upon nowadays as an anachronism. It is large, expensive, requires attention, often leaks, but it uses very little power and, in the long run, its upkeep is not always greater than that of a centrifugal pump.

For sticky or corrosive liquids the diaphragm pump is excellent, as the reciprocating ram or piston works in water and any leaks are simply water leaks.

Centrifugal pumps for boiler feed for high pressures are only satisfactory in large sizes. The boiler feed pumps in the author's factory are extravagant and troublesome. At full load the pump takes 155 kW, at no load it takes

102 kW. When originally installed the pumps gave a pressure of nearly 900 psi to feed boilers working at 650 psi. Such a margin was thought to be excessive so one stage was taken out, leaving 6 stages instead of the original 7. This gave a permanent saving of about 30 kW but only just came off. When the pump is nearly due for overhaul the boiler pressure has to be dropped to about 630/620 psi in order that the emasculated pump can feed at full load.

707. PROCESS MACHINE SPEEDS. There are many machines in process factories where the speed seems to have been decided upon by one of two methods : think of a number, double it ; or, add 50 per cent. for progress to the speed grandfather used.

Many machines do not call for any clear cut definite speed ; such machines, for example, as stirrers, mixers, rotary kilns, rotary driers, washing machines, etc. Their speed is largely a matter of personal opinion, and if the plant works all right it is left alone. Now there must be one particular speed at which every single machine will work best. If this speed is slower than that in use, power will be saved. If this speed is faster than the speed in use, output either by quantity or quality will be improved. It is therefore desirable that the best speed for each machine should be found by systematic trial. This is often impossible in the case of direct driven machines, but where speed variation is possible it should be tried.

Elevators, conveyors, etc., do call for a certain speed—the speed that will just carry the maximum peak load. Most elevators and conveyors, however, run far too fast to be on the " safe " side.

Not only is power saved by running machines slower, but noise and vibration are minimised, breakdown and maintenance are lessened.

708. EFFECT OF SPEED ON LOAD. In order to determine the effect of speed on the general factory load an experiment was made in the summer of 1940 in the author's factory during a period of light load. The governor of the turbine was adjusted at intervals so that the frequency of the supply was reduced by 1 cycle every 5 minutes, the voltage being reduced proportionately to the frequency. The results were not very consistent but they nevertheless showed a very definite trend. Fortunately the most important reading, the turbine steam consumption, was fairly consistent. The following figures were recorded :—

Cycles	*Volts*	*Amperes*	*Power Factor*	*kW*	*Steam Flow*	*Marginal Steam*
50	440	3700	·86	2300	88,000	57,400
49	430	3600	·82	2200	81,000	50,400
48	422	3800	·80	2000	80,000	49,400
47	414	3700	·74	1800	78,000	47,400
46	405	3800	·70	1700	72,000	41,400
45	397	4000	·60	1800	70,000	39,400

The test could not be carried below 45 cycles as pumps started to fail, by not reaching their tanks, motors started to trip out, etc.

The fixed steam consumption of this turbine is approximately 360 × Absolute Back Pressure, or, at the time of this trial, 360 × 85 = 30,600 lb./hr. By deducting this from the total steam flow to the turbine we get the actual flow that was required to run the factory at the various frequencies. It will be seen that the marginal steam consumption drops far more rapidly than directly as, or even as the square of, the frequency. About 45 per cent. of the factory load is made up of fans and centrifugal pumps, some of them with free discharge, but the majority with throttled discharge. It is therefore not at all clear why such a very large saving in steam was secured. It might be that a few centrifugal pumps failed to overcome their hydrostatic head unnoticed and therefore stopped pumping, but these must have been very few, their churning load would have been considerable, and such failure would not have occurred until say 48 cycles had been left behind.

Whatever the explanation, the fact remained that there was a great saving by running the plant slower.

709. METHODS OF SPEED REDUCTION. The general reduction of frequency had to be ruled out because many machines in the factory determine the quality of the product by virtue of their particular speed, and it would have been impossible to have increased the speed of these particular machines. It was therefore decided to do as much local speed reduction as possible to those machines that could stand it.

All belt- or chain-driven machines were tackled, slower speed motors were tried, many machines were connected to the 43 cycle supply that the factory possesses to run a battery of machines that must run at 1,250 r.p.m. Every effort was made to slow each machine down to the lowest point at which it would do its work. The result was that some 600 kW was saved in this way.

Together with saving made by pump stage removal the total normal day load was reduced from over 3300 kW to 2700 kW.

Hydraulic couplings are sometimes very convenient devices for speed variation with constant speed motor drive, but they are great power wasters. An hydraulic coupling is simply a liquid-cooled slipping clutch. No one would dream of using a mechanical slipping clutch for speed variation because the power wasted in slip would quickly assert itself by burning out the clutch faces. But in the hydraulic coupling the liquid takes the slip heat out of the clutch and dissipates it odourlessly and invisibly in the oil cooler. Of the power put into the induced draught fan motors in the boiler house at the author's factory, some 23 per cent. is dissipated in the hydraulic coupling oil cooler. Variable speed induction motors are simply air-cooled magnetic slipping clutches and the power is wasted just as surely in rotor resistance as by an hydraulic coupling or a friction clutch. Hydraulic couplings may be extremely valuable as shock absorbers.

710. OUTPUT AND SPEED. Reduced machine speed does not necessarily mean reduced output. Sometimes an increased or improved output results.

Sugar packeting machines running at 61 packets per minute gave a better output than at 64 due to less waste, fewer hold-ups, less strain on the operators.

A battery of stamping presses was driven fairly fast, the operators engaging the press clutches for each stroke. By greatly reducing the speed of the drive the operators were able to keep the clutch engaged and feed continuously with better output, less operator strain, less clutch wear and less power.

Process plant such as stirrers, rotary kilns, tumblers, etc., will often give as good an output with as good a quality at reduced speed. The speed of a machine is only too often somebody's guess, and a new guess may be just as good or better. Lower speed may give a better output or a better product—on the other hand it may not. The only way to find out is to try or to ask someone who has tried under identical conditions.

711. DIRECT COUPLED MOTORS. Where motors drive through belt, V-rope or chain, it is easy to make speed alterations. Where the drive is direct it is much more difficult.

D.C. drive provides a fairly economical speed adjustment, but it is only economical over a relatively small range. A variable speed motor is large and expensive, as is its regulator.

A.C. variable speed commutator motors are now available with a very large speed range, but they are very expensive.

Rewinding motors to give them an extra pair of poles may sometimes be economic. The power of the motor will be reduced, the rewinding is expensive and the new speed may not be satisfactory. Anyhow, by this means the speed variation can only take place in big jumps.

712. FREQUENCY CHANGING. Operating a group of machines at a lower frequency is sometimes an economic proposition, more especially if there are certain machines that will give a better result at a special speed.

If the factory generates its own power it may be possible on Saturday or Sunday to run the generating plant at low speed and to try the effect. If this looks really promising it may be worth while buying a small generator and driving it by V-rope, belt or chain at the reduced speed. It is not by any means necessary to use a proper direct-coupled synchronous frequency changer in every case. The lower frequency generator can often be driven quite satisfactorily by an induction motor. If the factory power plant consists of several reciprocating engines it may be possible to run one at a lower speed and feed its current into a special circuit.

In the author's factory frequency changers are used to get certain speeds that are technically necessary and that are unobtainable from a 50-cycle supply. Fortunately these frequency changers are lightly loaded with their special machines and these special frequencies are available in certain parts of the works for other speed reductions which have resulted in great economies. The approximate speeds available are given at the top of the next page.

Any induction motor can be used interchangeably on any of these supplies.

It may not pay to install frequency changers on a large scale simply to save power. But if there is a good case for the direct drive of certain machines at special speeds it may pay to install oversize frequency changers and thus have a ready means of getting savings on other machines.

No. of poles	*58 cycles 500 volts*	*50 cycles 440 volts*	*43 cycles 375 volts*	*25 cycles 220 volts*
2	3,300	2,900	2,490	1,450
4	1,650	1,450	1,240	725
6	1,100	970	830	485
8	825	725	620	360
10	660	580	495	290
12	550	485	415	240

713. INEFFICIENCY OF MACHINES. The fundamental fault that is the prime cause of the power problem is that most machines are very inefficient, often quite deplorably inefficient.

Machines that are largely cam-operated usually show that the designer has taken some pains to make the working part of the cam contour of the correct shape, but has taken no pains whatever with the return cam contour, his only concern being to get the return movement over and done with as quickly as possible. If cams are designed so that the whole of the movement is as gentle as it is possible to make it, not only is there often a very great power saving, but there is much less wear and tear on the machine parts.

Many machines take almost as much power running light as on full load—for example looms. Here is an assortment of readings from machines in the author's factory :—

Machine	*Output or speed per minute*	*Load* *Machine running on no-load*	*Machine on full load*	*Efficiency per cent.*
Sugar packeting machine..	61 packets	3·6	3·8	5
Can body maker	73 bodies	1·4	1·6	13
Can double seamer ..	290 cans	4·9	6·9	29
Rotary sugar drier ..	4 cwt.	5·0	7·5	33
Boiler feed pump ..	2,940 revs.	170	250	32
Sundry machine tools in engineers' shop ..	—	8·4	9·6	13
Average of all centrifugal pumps on liquor ..	—	—	—	21

So that apart from any electrical considerations there are tremendous possibilities from the mechanical point of view for power saving. Do we ever see advertisements saying that the advertiser's machine takes only half the power of those of his competitor ?

714. BAD LAY-OUT. Often plant is added to meet extra demand without adequate thought. The urgent consideration is simply to find an empty space in which the new plant can be dumped and to get it going as soon as possible. The original lay-out has probably been well thought out and all economies of

handling and driving have been considered, but later additions often involve wasteful handling and awkward drives.

A pumping system in the author's factory, which had been installed as a makeshift and allowed to carry on as a permanency, was found to have an overall efficiency of 1·6 per cent. ! It was a centrifugal pump of too large a capacity running too fast and discharging into too small a pipe.

Material is often allowed to drop unnecessarily far, when it has to be elevated, lifted or pumped up again. Gravity can be a good power-saving ally and must not be wasted.

715. LUBRICATION. The choice of lubricant is much more important than is generally realised. It is not so important to choose the maker of the oil or grease as the correct type of oil or grease. There is little to choose between all the first-class oil makes. It is very difficult to get really good advice. The maker of the machine can sèldom give real help. The experienced user is much more likely to know. Therefore it is probably better to find a user of the same type of machine and see whether any information as to the best lubricant can be extracted. The maker does not know unless a user has told him. All the maker can do is to specify a first-class make and make a guess as to the best type of oil.

The can double seamer tabulated in Section 713 is an excellent example of poor lubrication technique. A great tangle of fast running epicyclic gears is housed in the six-head rotating turret. Oil cannot be used lest it leak out and drop into a can, so that the turret is lubricated with grease. With certain types of grease the turret gets so hot that the hand can hardly be held on it, whereas with other types of grease it only gets about blood heat. Much of the load on the motor is due to churning this grease and shearing many grease films. The machine will not take load in the morning unless it has first run light for 20 minutes or so in order to soften the grease.

The textile industry have carried out investigations into the lubrication of spinning frames and have found that very marked power savings can be got from the correct choice of lubricant.

The substitution of ball or roller bearings for plain bearings, particularly on line shafts is another great power saver. But in many cases there would be no need to go to this expense if only the bearings were correctly lined up. Cases have been met where the power needed to drive a line shaft has been more than halved by lining up the bearings correctly.

716. IDLE RUNNING. One of the simplest methods (in theory) of saving power is by cutting out idle running. It is not so very simple in practice. It either calls for constant supervision or it must be done automatically.

A foot switch so placed that if the operator leaves the machine its motor trips can sometimes be used.

Some factories consider that idle running is of little importance as the machine is on no-load. The list of machines and their no load power consumption in Section 713 should dispel this idea. Only too often the idle no-load current is little different from the full load current.

During 1944 an attempt was made in the author's factory to measure the idle load. At the time of the test the load was varying between about 2,600 and 2,800 kW. Calculation gave 420 kW as the actual work done.

One Sunday, with no material in process, as much of the plant in the factory as possible was run idle under conditions as nearly as possible simulating real working conditions. Fans could not, of course, be run light.

The power needed to turn the idle wheels was 2,120 kW.

Work done (calculated)	420 kW	
Fans (estimated)	150	
Idle current (measured) ..	2,120	
	2,690 kW	
Average load		2,700 kW

This is a very remarkable agreement, and confirms the awful truth that this factory wastes 4/5 of its power in useless friction.

717. BLUNT TOOLS. In all machining operations sharp tools are of greater importance than almost anything else. Only a sharp tool will give a good finish to correct size. A blunt tool can easily use up to eight times the power taken by a sharp tool. The heat produced by a blunt tool (clear indication of power waste) accelerates blunting, and may cause distortion of the work with consequent uneven machining. A sharp tool will always run cooler or faster (or both) than a blunt tool.

It will generally pay to stop a machine that is running with a blunt tool, and sharpen or change the tool. The time lost in sharpening is more than regained in quicker, more accurate, better finished work and a substantial power saving.

718. LIGHTING. The march of civilisation as played by the factory inspectorate band is calling for greatly improved standards of factory lighting. This, on the face of it, would seem to demand a bigger lighting current. It may however be possible, by careful arrangement, to get greatly improved lighting from the same load.

Much can be done by correct placing of the lighting points and correct choice of bulb size. Three 100-watt bulbs give much better light distribution than one 300-watt. Big powerful bulbs are more susceptible to vibration than smaller bulbs, but if fitted where they are not subject to vibration they seem to have just as long a life. The more powerful gas-filled bulbs give rather more illumination per watt than the smaller bulbs which are also relatively very expensive. Fig. 398 indicates that the 200-watt bulb is less effective than 100-watt or 300-watt bulbs.

White or near-white paint is a great help. There is no reason why plant should be painted dark grey or why machines should be painted black. Hot plant should be painted aluminium. Other plant can be painted white, pale grey or deep sky blue. The addition of a little touch of bright contrasting colour

often gives a great illusion of brightness ; for example, plant painted pale grey and picked out on flanges and other points with scarlet looks very gay and appears much lighter than it really is.

One of the stock objections to the use of bright or light paint on machines is that oil leaks make dirty brown stains. By using white or nearly white paint there is born an urge to eliminate oil leaks. The author has seen a machine shop where all the lathes, millers, etc., were painted cream. It looked amazingly bright and cheerful, and oil stains were noticeably absent. A foundry has been painted cream, picked out in green ; it was very effective and lasted well.

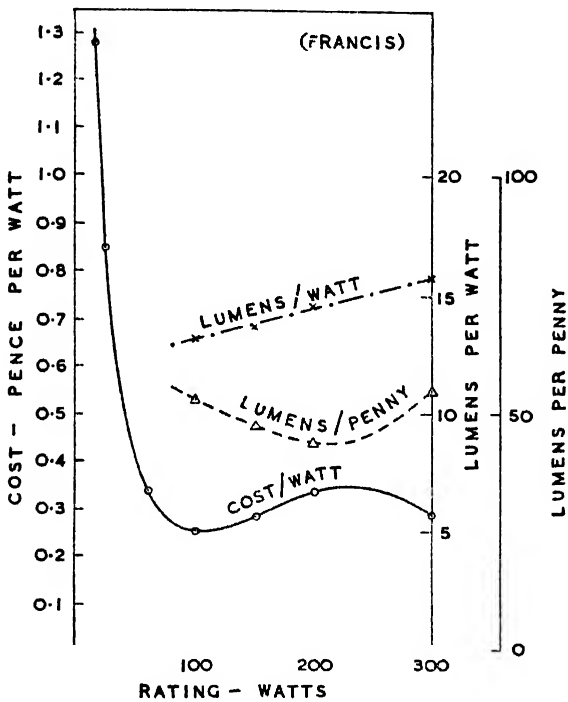


FIG. 398. COST AND LUMINOSITY OF ELECTRIC LIGHT BULBS (1945)

Flourescent lighting may possibly sweep all other lighting away, except for purely spot illumination. The current consumption for a given lighting is much less than with filament lighting. Whether the saving will warrant the cost of changing over depends on the local cost of electricity. In any new plant flourescent tubes should certainly be considered.

When estimating the saving by the use of fluorescent tubes the light intensity given by the makers must not be used. This is the new light output. It falls off with use.

The light from a fluorescent discharge tube is actually in the form of rapid short flashes which are separated by such a short interval that they merge together when seen by the human eye and are not normally detected. Moving

objects however show to many people, but not to all, a definite stroboscopic or flickering effect, which is very irritating and tiring to those who notice it. This stroboscopic effect can be greatly reduced by grouping the tubes in threes and running one off each phase.

*   *   *

CHAPTER 23

MAKING THE MOST OF ECONOMISERS

Men do not realise how great a revenue economy is.
CICERO. *Paradoxa.* B.C. 50

AN economiser is usually looked upon as simply an extension to the boiler. There is no rational justification for this view. The economiser need have nothing whatever to do with the boiler except to share the same fireside. Big steam savings can often be secured by modifying the conditions under which an economiser works.

719. WIDENING THE SCOPE OF ECONOMISERS. Feed water cannot claim the monopolistic use of an economiser. Anything which needs heating and can flow through an economiser without danger of damage from overheating can be heated in an economiser.

Many economisers which have been condemned by the insurance company for working at boiler pressure, or rather at the pressure for which they were installed, can often be brought back into useful service with a little rearrangement.

Many economisers are only partly pulling their weight because conditions have changed and they must be partly bye-passed to avoid condensation and corrosion, or water hammer.

Many boiler installations fitted with good economisers put their flue gases up the chimney considerably hotter than is necessary. In such cases it is often possible to fit additional or sub-economisers which can collect much of this useful wasted heat.

Why should a steam boiler be generally considered to have the sole right to an economiser ? Any furnace gases that are over 400° F., whether from a furnace, retort, oven, kiln or what not, should be considered in case an economiser can be fitted to do some useful heat abstraction.

720. WHAT AN ORTHODOX ECONOMISER IS AND DOES. The temperature of water in a boiler is approximately the boiling point at the particular working pressure, i.e., 338° F. at 100 psi, 388° F. at 200 psi, 489° F. at 600 psi and so on. Heat transfer from hot gases to steel boiler is relatively poor and requires a wide temperature difference to be effective. It follows that the gases must leave the boiler considerably hotter than the boiler temperature (several hundred degrees hotter).

The feed water usually goes to the boiler much cooler, but if this cool feed water is pumped through an economiser on its way to the boiler it can absorb much of the heat remaining in the flue gases.

The economiser is simply a bank or series of tubes placed in the flue at the back of the boiler.

There is an important difference between the heating surface of an economiser and that of a boiler. The water in the boiler and the boiler shell itself are at approximately the same temperature throughout. At the furnace end of the boiler there is a very large temperature difference between the gases and

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the boiler and the heat transfer is good and quick. As the gases pass through the boiler they get cooler and the heat transferred from them to the boiler decreases. It follows, therefore, that each succeeding attempt to reduce the temperature of the exit gases by increasing the boiler heating surface will give a diminishing return.

The economiser, however, has a channel of water which increases its temperature all the way, so that obviously both the water and the gases have a temperature gradient. By arranging the gases and the water to flow in opposite directions the heating surface of the economiser can become equally effective at the gas outlet as at the hot end (within certain limits). For this reason additional heating surface added to a boiler plant to extract more heat from the flue gases is cheaper and smaller if it takes economiser form than as additional boiler heating surface. In fact it is impossible to cool the gases adequately in the boiler, even by extending the surface beyond practical limits, whereas an economiser can cool them by any reasonable amount.

There are limits to the use and size of an economiser—one at the hot end, another at the cool end.

721. THE LIMIT AT THE HOT END. In the original type of vertical cast iron economiser (i.e. the type still commonly found in existing plants and to which this chapter is mainly intended to refer), water should not be raised to the full temperature of the boiler, otherwise steam will be generated where there is no provision to deal with it. In practice it is usual to proportion the economiser so that the water temperature at the outlet is not raised to within 50° F. of the boiler temperature. This is to avoid water hammer due to the sudden condensation of pockets of steam which would otherwise be formed in the economiser, under fluctuating load conditions.

Methods of dealing with water hammer troubles are described in this chapter.

In some modern plants steam formed in the economiser can pass to the boiler without causing trouble. With these "steaming economisers", the limit at the hot end is practically eliminated.

722. THE LIMIT AT THE COOL END. The water should not enter the economiser below a certain temperature otherwise rapid corrosion of the tubes will take place.

Before coal can be burnt, the moisture contained in it must be evaporated. All the hydrogen in the fuel burns to water vapour so that flue gas from the burning of coal contains an appreciable percentage of water vapour. Below a certain temperature this water vapour will condense—this temperature is called the "Dew Point".

If cold water is fed into an economiser, water vapour from the flue gas will condense on the outside of the cool tubes. This would not matter very much if the condensed water was pure, but it is not; it always contains acids, probably a mixture of sulphurous and sulphuric; these result from the burning of some of the sulphur which is present in all coals to a greater or lesser degree. The presence of sulphuric acid raises the dew point and is thus doubly undesirable.

723. ACID FLUE GASES. Sulphur in coal burns normally to sulphur dioxide, but traces of sulphur trioxide are often found in flue gas. It has been suggested that the flue dust acts as catalyst to cause the SO_2 to take up oxygen to form SO_3. Any SO_3 present combines with the condensed moisture on a cool tube and forms sulphuric acid which attacks the metal, forming ferric sulphate. Ferric sulphate is a powerful catalyst for inducing SO_2 to oxidise to SO_3. So that a minute amount of SO_3 in the gas can cause the formation of an appreciable amount of sulphuric acid on the tubes.

The acid thus formed can cause rapid corrosion of the economiser tubes. It is therefore common practice to preheat the water entering the economiser so as to preclude, or at any rate minimise, condensation and corrosion.

724. HISTORICAL DIGRESSION. Nowadays dew point is a big, bad bogey. Two generations ago it was no great shakes. Why ? Sixty years ago the gases left the boiler very hot and much good heat went up the chimney. Green invented the economiser and what did he do ? He made his tubes of heavy cast iron and dew point troubles were seldom heard of in the good old days. Green also scraped the tubes continuously to keep them clean.

Boiler pressures increased and cast iron was superseded by steel. Cast iron is extremely resistant to corrosion but steel likes nothing better than to corrode in any convenient acid. Dew point became of great importance with steel economisers and corrosion a real danger. So the big high pressure stations had to heat their feed water before it entered the economiser. This they did by steam bled off from the low pressure end of their turbines. What they lost on the economiser swings they more than recovered on their turbine roundabouts because bleed heating was a first-rate thermodynamical improvement.

So the big high-pressure power stations rushed gaily into multi-stage bleed feed heating, in order to cut the maximum possible corner off the entropy diagram, until they had so heated up the feed water that the economiser could not cool the gases down sufficiently. So air heaters had to be installed to absorb the heat previously taken up by the economiser. Most air heaters are flimsy steel affairs which suffer even worse than steel economisers from dew point corrosion troubles because the air intake end of an air heater is often really cold.

These tiresome tendencies are being corrected by two trends which are at present noticeable ; a somewhat less exuberant use of bleed heating and the development of cast iron air heaters.

725. DEW POINT. Condensation will take place on a cool heating surface if the surface is at a temperature below the dew point of the gas. The temperature of the flue gas has nothing to do with condensation. That is to say, if a particular flue gas has a dew point of 100° F. and a temperature of 1,000° F. its moisture will condense on an economiser tube if the tube surface has a temperature of 95° F. or so. It is often loosely said that flue gases must not be cooled down to their dew point. The bulk temperature of the gas is no guide at all. Locally the gas is bound to be cooled to below the dew point if it is in contact with a surface that is cooler than the dew point.

What is the dew point of flue gas ? It all depends on the fuel and the way it is burned, and it is difficult and indeed dangerous to try to lay down limiting figures. For what they are worth the following are rough guides and apply to dry fuel burnt with dry air :—

TABLE LXXIV. APPROXIMATE DEW POINTS OF FLUE GASES

FUEL	DEW POINT
Blast furnace gas	55° F. to 60° F.
Producer gas—Coke	85° F.
Producer gas—Coal	140° F.
Town gas	140° F. to 150° F.
Coal—Good Midland	80° F. to 150° F.*
Coal—Bad Sulphurous	Up to 300° F.

*These figures must not be accepted as authoritative. Seemingly good coals sometimes give dew points of over 250° F.

726. CORROSION—DOES IT MATTER? If the surface is steel—yes. Steel corrodes readily and rapidly and is generally of light section, so that corrosion must not be allowed. Cast iron is another matter. It corrodes slowly and there is usually a generous margin of metal.

An economiser generally pays for itself in three years—often in less. With present high-priced coal it is probably much less if worked on three shifts. Therefore even if the economiser corrodes away in three years no money has been lost—or saved. Instead of buying weekly coal we buy three-yearly economiser. Anything over three years' life is pure net gain.

It is curious that this point is not fully appreciated by owners who regularly renew their grates every few years without complaint.

727. WIDENING THE LIMITS. Now, what can be done to extend the limits that restrict the gain from the economiser ? At present in only too many plants the restrictions are accepted and the limits upheld by bye-passing the flue gases to prevent water hammer and by pre-heating the feed water, only too often, by means of live steam. Both these lamentable heat-wasting practices can be obviated quite simply. Some of the methods are shown in the following set of examples.

For these examples an ordinary "Lancashire" boiler plant, with an economiser of vertical tubes in parallel, has been chosen. Reasonable working temperatures are given, but the actual temperatures obtained must depend of course on the quality of the fuel burned, the quantity of steam evaporated, the quantity of flue gases, the amount of excess air, the draught available, etc. ; so that the temperatures given can only be taken as indications of what can be done.

While considering the examples the following rough rules are helpful :—

For 1° F. rise in feed water by the economiser, the flue gas temperature is lowered 2° F. (See Section 577.)

Approximately every 10° F. increase in feed water temperature from the economiser or by the use of waste heat saves 1 per cent. of fuel.

728. CURING WATER HAMMER DUE TO HIGH FEED TEMPERATURE. A factory raises steam for process work at 150 psi. Originally only 60 per cent. of the condensate was recovered at 210° F. ; 40 per cent. of treated make-up water was added at 60° F., giving a feed temperature of 150° F. at the economiser inlet, and 320° F. at the outlet. See Fig. 399.

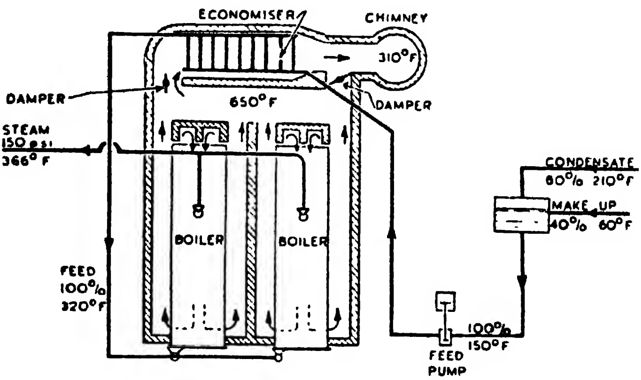


FIG. 399.

A fuel economy drive rightly gathered in more valuable condensate ; this resulted in the collection as condensate of 85 per cent. of the boiler feed and the increase in feed temperature to 187° F. These conditions are shown in Fig. 400.

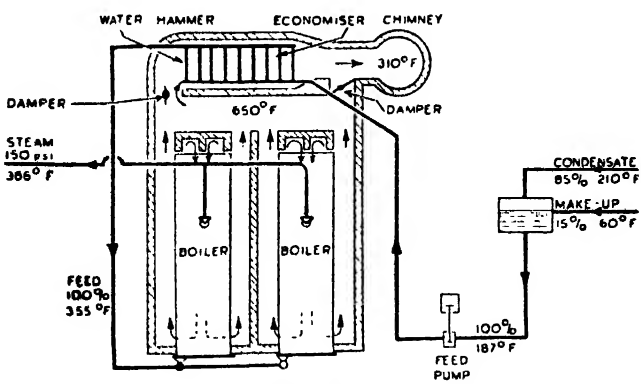


FIG. 400.

The water then left the economiser at only 11° F. below the steaming temperature and, as this margin was insufficient to cover normal variations in

running conditions, ominous water hammer occurred. To get over this the setting of the economiser dampers was changed and, to be on the safe side, they were locked so as to bye-pass rather too much gas to the chimney.

The new condition is shown in Fig. 401 where it will be seen that the chimney gas temperature has increased to 410° F. and the economiser water outlet temperature lowered to 305° F.

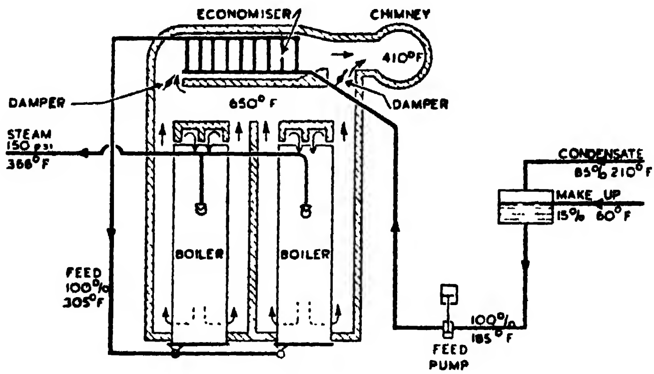


FIG. 401.

The management soon realised that, although they had increased efficiency about 3½ per cent. by recovering the condensate, they were now also reducing it 4 per cent. in the economiser (the net result being ½ per cent. loss) and other tactics were considered to prevent water hammer without bye-passing the gases.

The final decision was to revert to the original setting of dampers and run the feed pump continuously at full speed, bleeding a portion of the feed water to a flash tank feeding steam to the process main at 5 psi. See Fig. 402.

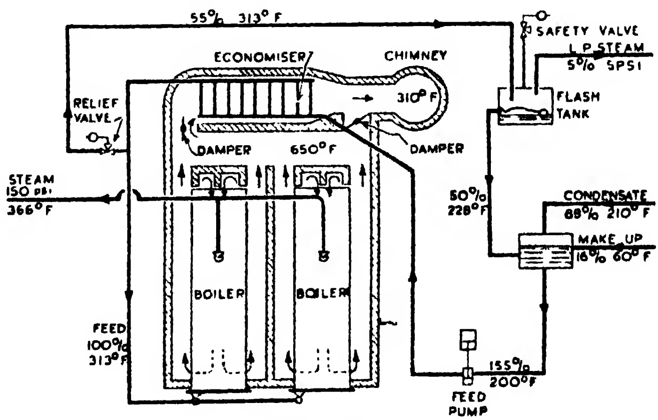


FIG. 402.

A relief valve was inserted between the main feed line and the flash tank to maintain the required pressure on the feed line. The boiler feed valves

controlled the amount of water fed to the boiler and all surplus water passed automatically to the flash tank where the released sensible heat was sufficient to flash about 5 per cent. of the boiler feed into steam at 5 psi.

Adequate feed pump capacity is needed in this arrangement and the installation of an additional pump may have to be considered, and may be worth installing.

N.B.—Before operating such an arrangement the boiler insurance company should be consulted if it is found necessary to set the relief valve above normal blow-off pressure.

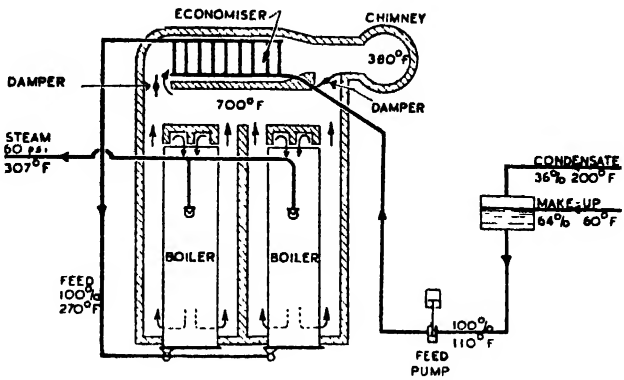


FIG. 403.

729. CURING CONDENSATION DUE TO COLD FEED WATER.

A factory raises steam at 60 psi. 36 per cent. of the water evaporated is recovered as condensate, the remainder being made up with treated water. The arrangement is shown in Fig. 403.

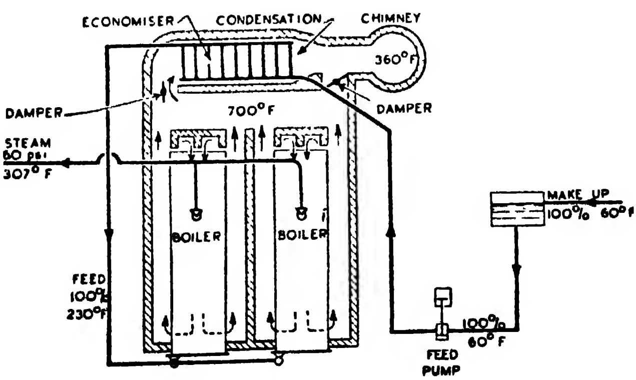


FIG. 404.

The management decided to use all the condensate for a process that needed distilled water ; therefore, the boilers were fed entirely from the cold treated water. These conditions will be seen in Fig. 404.

The result was severe corrosion of the economiser due to condensation of the flue gases. After consideration, the management decided to make two additions to the feed circuit. See Fig. 405. A coil was installed in the feed tank to recover the latent heat in the exhaust steam from the feed pump. The feed temperature was raised approximately 10° F. by this. This was a useful step in the right direction, but was insufficient to cure the condensation, for which an economiser inlet water temperature of 110° F. was to be aimed at.

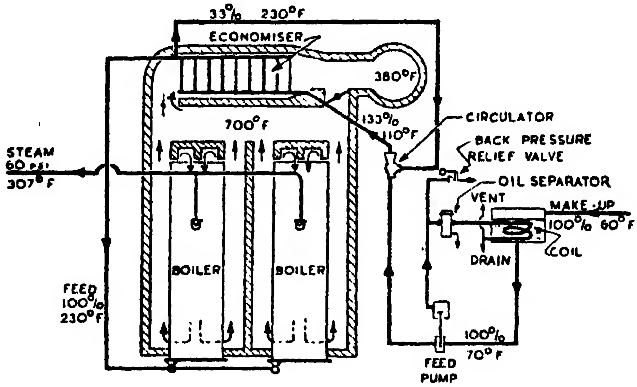


FIG. 405.

By recirculating approximately 33 per cent. of the feed water around the economiser, the inlet would be raised to this temperature and this was done by the installation of a "National circulator".

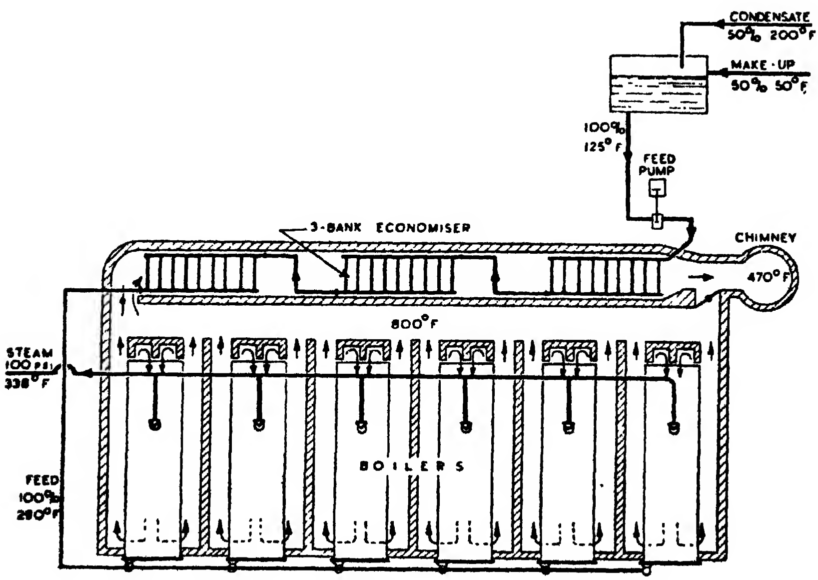


FIG. 406.

There are several ejector type "circulator" fittings on the market, but recirculation can also be done by centrifugal pump. Sometimes it is sufficient merely to "short circuit" some water from the economiser outlet to the feed pump suction line by means of a suitably valved pipe connection. The water circulators work on exactly the same principle as the steam circulators described in Sections 501 and 503.

730. ECONOMISER BECAME TOO BIG. A distillery with six boilers operating at 100 psi recovered 50 per cent. of the condensate for boiler feed at 200° F. The remaining 50 per cent. was made up with soft well water. One large three-bank economiser was common to all six boilers, as shown in Fig. 406.

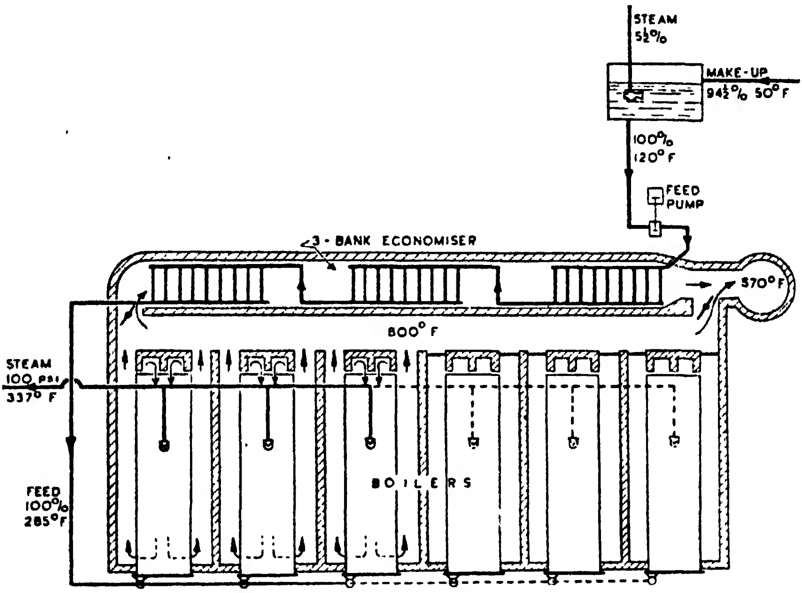


FIG. 407.

As the result of a great economy drive, the process was turned inside out. As much condensate was recovered as possible and all of it was used in the process. At the same time the steam demand was so reduced that three boilers could now carry the load. The well water was too cold, of course, to put into the economiser and was warmed to 120° F. by means of live steam injected into the feed tank. The economiser was now far too large for the smaller amount of feed water passed through it, so that a large proportion of gases had to be passed direct to the chimney. This deplorable arrangement is shown in Fig. 407.

The management, being economy-minded, "felt their position keenly", so they reorganised their economiser, effecting a remarkable improvement. Fig. 408 shows what they did.

The three banks of the economiser were separated and given three separate jobs to do. The bank taking the hottest flue gases was used as a normal economiser heating the boiler feed. The middle bank was used to pre-heat the

cold well water with rapid circulation so as to prevent condensation troubles. The pre-heated water to the first bank was taken from a mixing tank through which water from the middle bank was continuously recirculated. The amount of water passing through the circulating pump was just over twice the amount of feed water. The bank nearest the chimney, where the gases are coolest, was connected to the low pressure hot water heating system warming the office block, thus enabling an independent coke fired boiler to be shut down.

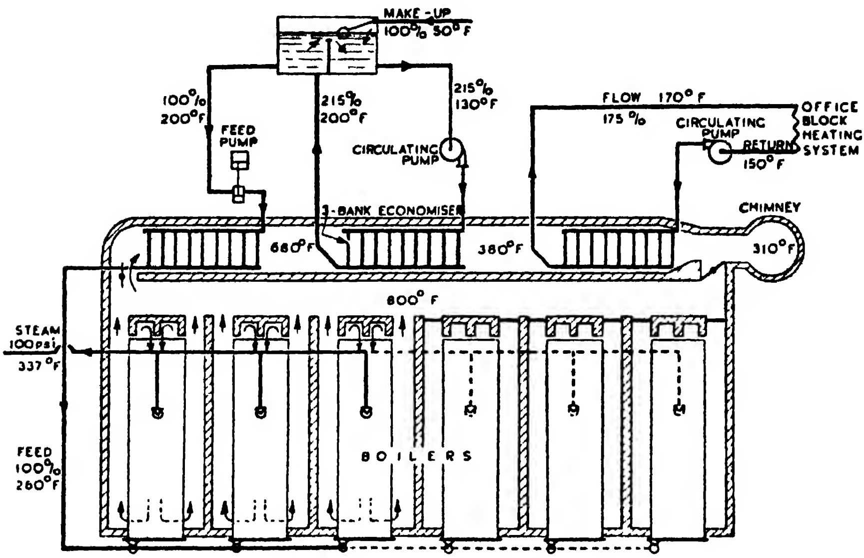


FIG. 408.

It will be noticed that after the alterations only the first bank of the economiser was working under pressure. When the space heating economiser is not wanted in summer, a length of pipe is taken out in both flow and return, as near the economiser as possible, and the economiser is drained ; the temperatures are too low to do it any harm.

731. THE USE OF A CONDEMNED ECONOMISER. A factory with an old economiser, which brought the flue gas temperature down to 500° F., was told by its insurance company that the economiser could not be re-insured for the required boiler pressure. A new economiser was erected, therefore, alongside the existing one. When the new economiser was installed, the flues on the old plant were so altered that the gases leaving the new economiser at 500° F. passed through the old one. The old economiser was split into two parts, the larger hot part was used with a circulating device to heat water for process work and the remainder was connected to the pumped hot water system dealing with the space heating of office and warehouse. As both sections of the old economiser then worked at a very low pressure, they were good for many years' useful service.

732. ANOTHER CONDEMNED ECONOMISER. In another works a large economiser installation was condemned and, due to the way in which it was situated, it would have been a tremendous job to replace it. Further, it would have been impossible to replace it without interfering with production for a long period. The outlet water temperature from the economiser was only 215° F. so it was left in position and the system was changed to operate it at low pressure. The outlet discharged into a closed insulated tank provided with a safety valve set at 5 psi. The water was drawn from the tank to the boilers by the boiler feed pumps. The system worked very well and the old economiser was given a new lease of life. It was estimated that the cost of replacement would have been something like £20,000, whereas the alteration cost less than £2,000.

733. USING AN ECONOMISER AS AN AIR HEATER. The economiser in a tannery boiler plant was condemned. It was converted into an air heater for heating air for the drying rooms. It has been working satisfactorily in this way for 25 years.

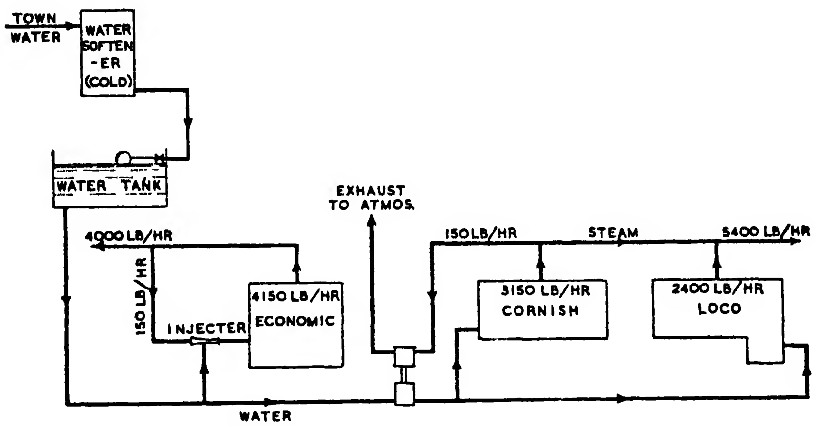


FIG. 409. LAUNDRY BOILER PLANT

734. OLD BOILER CONVERTED INTO ECONOMISER. In a laundry in Durham there were three boilers giving the following outputs :—

	lb./hr.
Economic	4,150
Loco	2,400
Cornish	3,150
	9,700

The feed water was quite cold to the boilers. The economic boiler was fed by an injector (see Fig. 409).

Many economies were made and the feed water was heated to a good temperature by means of exhaust steam. The steam demand was reduced to 8,350 lb./hr., which, now that the feed was hot, could be produced from the economic and loco boiler without using the Cornish boiler.

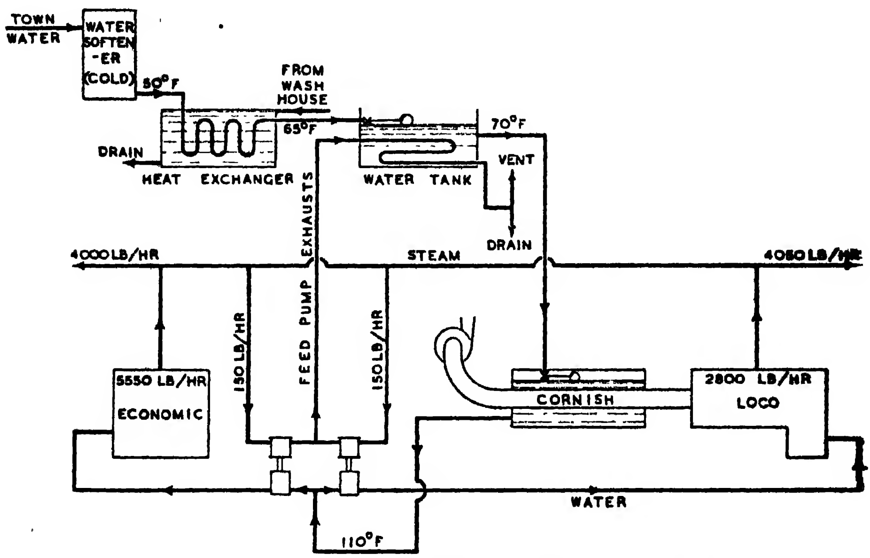


FIG. 410. OLD BOILER USED AS ECONOMISER IN LAUNDRY

The hot flue gases from the loco boiler were passed through the fire tube of the Cornish boiler which added 40° F. to all the laundry hot water, amounting to 28,960 lb./hr. (See Fig. 410.) Assume a boiler efficiency of 65 per cent.

This amounts to a coal saving of :—

$$\frac{28{,}960 \times 40}{10{,}000 \times \cdot 65} = \text{say } 178 \text{ lb. coal/hr.}$$

If the net working week is 45 hours, the annual coal saving is 175 tons worth £1,000.

735. WIDENING THE SCOPE OF ECONOMISERS. There is no reason why an economiser should be looked upon as solely a water heater. An economiser can be used for heating liquors, lye, wort, oil, tar—in fact anything that will flow through it and that requires heating.

Sufficient examples have now been given to show that there are all kinds of ways by which the scope of the economiser can be widened. The economiser can be used :—

1. As an orthodox high pressure feed-water heater.
2. As a low pressure pre-heater for cold feed-water.
3. As a process water heater.
4. As a heater for process liquids other than water.
5. As a space heating boiler.
6. As a water super-heater for flash steam production.

There are many economisers working to-day without troubles where the inlet water temperature is only 110° F., or with exit gas temperatures of 220° F.

736. SUPER-EFFICIENT POWER STATIONS. The present tendency in large boiler installations is to put the gases to the chimney at or near 300 °F. from the air heater. It is possible that an improvement in efficiency would be obtained by cutting out one of the low temperature turbine bleeds and substituting for its stage heater a sub-economiser between the air heater and the stack. If a station is working a high-grade technique and cuts out one of its low temperature bleeds it will drop in thermodynamical efficiency. If it circulates its condensate through a sub-economiser instead of a bleed heater and reduces its flue gases from say 300° F. to say 200° F. it will probably put 50° F. into its feed water thereby gaining on balance. The sub-economiser can be of cast iron as it has not got to withstand more than the pressure necessary to cause the water to flow through it.

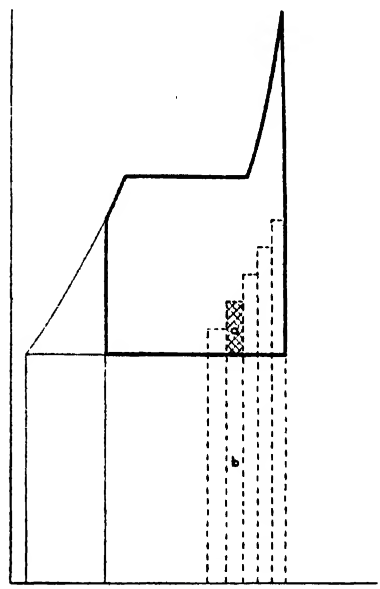


FIG. 411. SUB-ECONOMISER REPLACING TURBINE BLEED

Fig. 411 is the entropy diagram of a high performance station. If the fourth bleed, areas $a + b$, is dispensed with, those two areas will pass through the turbine. Area a will be useful work; area b will be rejected to the condenser. But if the heat, areas $a + b$, has been supplied by the sub-economiser, area a is free net gain and area b can be ignored. So that there is a net power gain of area a, which adds a good 1 per cent. on to the cycle efficiency.

737. KILNS, GLASS TANKS, FURNACES. Generally, serious efforts are made by all good plant owners to save as much heat as possible from the boiler flue gases, but it is regrettable that flue gases from other types of plant are treated in a much more off-hand manner. Some kilns and furnaces lend

themselves admirably to the application of an economiser; for example, the dew point of the flue gas from blast furnace gas is very low and much good heat could be extracted from it.

738. FEED TANKS OR HOT WELLS. In many factories cold water is deliberately added to the " Hot Well " because the feed pump will not handle the hot water. In other factories condensate is deliberately not collected for the same reason. Again, feed pump exhaust is often wasted, when it could well heat the feed water, because of feed pump trouble. The term " Hot Well " is a contradiction. If it is hot it cannot be a well. If it is a well it cannot be hot. If the feed water is to be very hot the tank must be above the feed pump suction. If a sunk tank is needed for condensate collection the hot water can be raised to a higher feed tank by means of a return or pumping trap. The trap exhaust can be led through a coil in the feed tank and give up its heat to the make-up water. Suitable pump suction heads are given in Table XLIV. If the boiler feed consists of condensate at or about 212° F. many managements or engineers appear to think there is no scope for an economiser. If the boiler pressure is only 100 psi the temperature of the water in the boiler will be 338° F. This means that about 75° F. can well be put into the feed water by an economiser, which will reduce the flue gases by about 150° F. Even at lower boiler pressures it may be an excellent plan to put in an economiser, part of which can do orthodox feed heating, while the remainder heats process water or does space heating.

739. SUB-ECONOMISERS. Because an economiser is fitted to a boiler or other plant it does not necessarily mean that the economiser is big enough or is doing all the possible heat recovery. It may prove practicable to add another economiser between the original economiser and the chimney. If the gases are leaving the orthodox economiser at 500° F. a sub-economiser can probably cool them down to 300° F. If the furnace combustion is good, without too much excess air, there will be at least 16 lb. of flue gas for every lb. of coal burnt. If one Lancashire boiler is burning 8 cwt. of coal an hour there will be a flow from the flue of over 14,000 lb. of flue gas per hour. If the flue gas can be reduced in temperature by 200° F. it will be possible to recover 700,000 Btu/hr.—just about the total space heating requirements of the brewery discussed in Sections 612 to 635.

740. POSSIBLE ECONOMISER PITFALLS. A word of warning—don't be deceived into thinking you have a very efficient plant or that there is no scope for a sub-economiser because there is a low gas temperature at the base of the chimney. Such a condition may be due to too much excess air or to air leakage through brick settings, etc., which should be periodically examined.

Another word of warning—if success crowns your efforts and you succeed in greatly lowering the temperature of the chimney gases, you may be in trouble with draught. A cool chimney gives a poor draught. Induced draught fans may be needed. These are well worth while and the power required will always cost less than the saving secured by cooling the gases. Sometimes draught fans as well as feed pumps can be steam driven with advantage, if the exhaust steam can be used from them all. The ease of speed control is useful.

CHAPTER 24

HOW TO SET ABOUT STEAM SAVING

The common problem, yours, mine, everyone's,
Is—not to fancy......but—finding first
What may be, then find how to make it fair
Up to our means.

ROBERT BROWNING. *Bishop Blougram's Apology.* 1885.

THIS chapter is largely composed of the author's opinions. There is no cut and dried scientific method of saving steam. The author can only claim justification for airing his views because he has been associated with a very large and somewhat spectacular steam saving campaign in a factory that was generally considered " efficient " before the campaign started.

741. THE THREE FUNDAMENTAL STEAM SAVERS. There are three fundamental things, and three only, that should guide our steam economy, and we should strive after them with might and main.

(1) Prevent the escape of heat.
(2) Reduce the work to be done.
(3) Use the heat over again.

In seeking to obey these three golden rules we can easily trip up. We must not go blindly collecting flash steam or waste vapour unless we are quite sure that we can use it and are not just going to waste an equal amount of live steam elsewhere. We must consider each of these rules in turn and in some detail.

742. PREVENTING THE ESCAPE OF HEAT. There are several types of escape and there are several kinds of heat. We will call one lot primary and the other secondary.

Primary Heat Losses :—

(1) Radiation. Sections 138, 161–170, 464.
(2) Leaks. Section 160.
(3) Safety Valve Blows. Sections 129, 424, 437, 524, 525.
(4) Faulty or Byepassed Traps. Sections 278, 314.

Secondary Heat Losses :—

(*a*) Engine exhaust.
(*b*) Flash steam.
(*c*) Condensate.
(*d*) Blowdown.
(*e*) Pan, vat and evaporator vapour.
(*f*) Hot waste effluent.

The primary heat losses can be tackled straight away. No plan is necessary. No balance need be taken out. Primary heat is virgin heat. Primary heat losses are the only ones about which there is no shadow of doubt whatever.

They should be stopped instantly. All the primary losses in the above list have already been dealt with in preceding sections, to which reference is made.

The savings obtained from eliminating the primary heat losses should not, in any self-respecting works, amount to very much, though safety valve losses may be very much larger than is believed. But they can be attacked instantly without thought or plan and the attention given to them will have a good tonic effect on the thermal morale of the factory.

The secondary heat losses present a very different problem. The possible saving may be very large indeed, but, unless a proper balance is taken out and a proper plan made, there may be no use for the heat that may have been saved at great trouble and expense. Most of the secondary heat losses hardly come under the heading of " preventing the escape of heat " ; they are often more correctly part of the technique of using the heat over again.

743. PRIMARY HEAT LOSSES. Everything hot should be lagged. It is much more important that everything should have some lagging than that some things should be elaborately lagged and other things left bare. Tables XXII and XXIII show this very clearly. If a certain amount of money is to be spent on lagging, 1 in. or even ½ in. applied everywhere is far better value than 1½ in. or 2 in. applied to certain privileged plant.

Lagging is an absolutely certain winner, but of course, as in other things, there must be commonsense and moderation. Lag warm things thinly and hot things thickly, but lag everything that is above the local " critical" temperature which should, if possible, be found out.

The " critical " temperature above which it pays to lag and below which it does not pay must vary with every factory. It depends on the return that the expenditure must give, and on the financial policy of the concern. It depends on the place and site of the factory, and its freedom or otherwise from draughts. It depends on the cost of coal. The subject has already been discussed in Sections 166 and 167, but a little repetition will do no harm.

Select as long a length as possible of bare pipe carrying the hottest water possible in such a position that the flow through the pipe can be exactly calculated or accurately measured. It may well be that there is no such pipe ; then the nearest approach to the conditions that is practicable should be made. With the help of Table XXIV find the area of the pipe. Measure the temperature of the water entering and leaving the pipe. Measure the temperature of the surrounding air. Find the amount of water flowing during the time the temperature measurements were taken. This will give the heat loss/sq. ft./hour. See that this figure is of the same order as the figures given in Table XXIII. By the method described in Section 350 find the mean temperature difference. This will give the loss/sq. ft./hour/° F. difference. By the method described in Section 166 find the coal value of the heat loss. Get the pipe lagged with good lagging of reasonable thickness—say 1 in. or 1½ in. Get the exact cost. Take the heat loss over the lagged pipe and thus

calculate the heat that has been saved. It is then easy to find the time in which the cost of lagging will be repaid. If the temperature drop was 100° F. and the repayment time was three years we shall not be very far wrong in saying that with a temperature drop of 60° F. the repayment time will be six years. If it be decided that lagging must repay its cost in six years, everything that is 60° F. hotter than its average surrounding air can and should be lagged. We must not take direct proportion for the heat loss and temperature difference because the hotter a thing is the greater is the rate of heat loss apart from the temperature difference. Lagging should be painted with aluminium paint.

If lagging is properly done and is properly covered and if the paint is renewed sufficiently often, lagging can easily have a life of 20 years or more.

Leaks need no comment or discussion. They have been dealt with in Chapter 4. They must be stopped entirely as soon as possible.

Safety valve blows are, as has been explained in Chapters 3 and 15, largely a matter of management. The wastefulness of certain types of safety valve have been discussed in Chapters 16 and 19. Safety valves merit close attention.

Traps must be properly maintained. A routine schedule of inspection and service is probably the only way in which traps can be kept in satisfactory working order If traps are byepassed when starting up high pressure plant from cold—and byepassing is the only safe rule—the management must see that the factory organisation is such that the byepasses are shut as soon as the plant is hot.

744. SECONDARY HEAT LOSSES. Most of the secondary heat losses come under the heading of using the heat over and over again, and will be discussed under that heading. It is generally unwise to tackle the secondary heat losses without taking out some kind of heat balance, be it never so simple and crude. It is quite useless just looking at the blast of steam from some pump exhaust or flash tank and collecting it by using it for heating water until it is certain that some other steam will not in consequence be blown to waste. An excellent example of this appears in the laundry illustrated in Section 605. The wasted flash steam from the collected condensate amounts to 280 Btu/lb. dry laundry, and the impetuous enthusiast might set about a flash collecting scheme, but what in the world can possibly be done with the flash heat ? Far more than enough heat for water heating is already being provided by the engine exhaust, so that the collection of the flash could only make more exhaust blow to atmosphere, no economy would be secured and all the cost of the flash collecting system would have been wasted.

Similarly there must be clear thinking regarding lagging. The lagging of a hot pipe is not necessarily saving primary heat. If the pipe is carrying condensate the heat is secondary. By lagging the condensate pipe the condensate may be so hot at the hot well that it must be cooled down before the pump can handle it. As has been shown in Section 398 it may pay to do the opposite of lagging to some condensate pipes, for example in space heating systems, namely fit them with gills or fins when there will be no flash heat to save.

So that one important thing is now clear. Before going baldheaded at lagging in order to save primary heat losses, make quite sure that they are really primary and not secondary.

745. REDUCING THE WORK TO BE DONE. This is a most fruitful source of saving and should be tackled before attacking the secondary heat losses. Sometimes it can be done without taking out the heat balance, but generally it is better to get the heat picture of the whole factory before investigating all the ways in which the work that the steam has to do can be reduced.

The kind of savings that can be made under this heading are :—

Reduction of processing temperature
Reduction of reprocessing
Quick processing
Full loading
Mechanical water removal
Processing at lower water content
Taking advantage of exothermic or endothermic reactions
Increased yield
Reduction of power load

Let us examine each of these in turn, except the last which has been dealt with in Chapter 22.

746. REDUCTION OF PROCESSING TEMPERATURES. Some of the benefits of process temperature reduction are obvious. If the processed material is being heated to a temperature higher than is really necessary some useless heat is being used. This heat will probably be lost, because it is often difficult or impossible to take the heat from the product by heat exchange. In addition, higher temperatures mean higher radiation losses from all the plant surfaces.

Apart from heat savings there are many products that deteriorate at high temperature and quite a big saving of material may be made by processing cooler—this is especially the case with organic substances.

A more subtle advantage from the use of lower temperature can be found in any sugar factory. When sugar is being crystallized from a syrup it is necessary to concentrate the mother syrup to a certain supersaturation. Now the saturation point varies with the temperature, that is to say, more sugar is dissolved in hot water than in cold water.

We will take two crystal batches each giving 10 tons of sugar crystals, but one batch will be crystallized at 175° F. and the other at 140° F., the mother syrup in both having the same supersaturation, namely 1·1 (which means that where 1 unit of sugar in solution represents saturation or all that the water will dissolve, there are actually 1·1 parts of sugar in solution due to over-concentration by the evaporation of water). We will assume a crystal yield of 50 per cent. and that the original liquor has an initial density of 68 per cent.

Input. 20 tons of sugar in a water solution of 68 per cent. density

68 tons of sugar in	32	tons of water
20 ,, ,, ,, ,,	9·41	,, ,, ,,

Density of syrup at 1·1 supersaturation	80·7 per cent. at 175° F.
	76·4 per cent. at 140° F.

Output (1). 80·7 tons sugar in 19·3 tons water
therefore 10 „ „ „ 2·39 „ „

Evaporation 9·41 − 2·39 = 7·02 tons of water.

Output (2) If 76·4 tons sugar in 23·6 tons water
10 „ „ „ 3·09 „ „

Evaporation 9·41 − 3·09 = 6·32 tons of water.

So that by finishing the crystallizing process at 140° F. instead of 175° F. we save the evaporation of 7·02 − 6·32 = ·7 tons of water per 10 tons of sugar. On an output of 12,000 tons a week this amounts to a weekly evaporation saving of 840 tons, or an annual coal saving of 5,500 tons.

747. REDUCTION OF REPROCESSING. Many products soiled in process are 100 per cent. recoverable. For example, laundry or sugar. There is consequently a tendency on the part of the operators to be very careless and allow much floor spillage, etc., because they know that the material is not lost but is recoverable.

The reprocessing due to all causes in the author's factory is normally between 1 and 2 per cent. It was only reduced to this figure after a long, weary campaign. It is satisfactorily maintained at this level by ordinary managerial control through the medium of a book in which the weight of reprocessed material from every department is entered each week. During the autumn of 1940 the book, amongst other things, was lost. The loss was not noticed until the summer of 1941 as the management had other preoccupations. In the meantime the reprocessing had risen to over 4 per cent. Without any other action whatever, the provision of a new book brought the reprocessing back to 2 per cent. within 6 weeks. The reduction of reprocessing by 2 per cent. was, at 1941 output, equivalent to a saving of nearly 14 tons of coal a week.

748. QUICK PROCESSING AND FULL LOADING. Heat loss by radiation and convection from tanks, vats, etc., accounts for much of the heat used in most factories. If the processing time can be halved, only little more than half the plant will be needed, and consequently the heat lost by the processes will be nearly halved.

The provision of large tanks is one of the easiest substitutes for smooth process operation. Tanks are very necessary, but often tank capacity is greatly overdone and material is kept in heat for hours longer than is really necessary. If there is ample tank capacity, hitches, hold-ups and surges in the process can be nicely hidden. If the process can be so organised as to flow smoothly it may be possible to put half the tanks out of action and save their heat loss completely.

One of the greatest sources of waste in factories using drying processes is underloading of the drier. Suppose a drying calender is running at a certain speed. The steam pressure is adjusted to give complete drying of the material in one pass through the calender at this speed. Now suppose the speed is such that the operator cannot feed the work sufficiently fast to keep the whole of the heating surface covered. If the machine is slowed down to such a speed that the operator can keep the calender fully loaded the work will be longer in contact with the heating surface and the steam pressure can be reduced. Four

benefits result. At the lower pressure the machine losses will be lower. Good uncovered heating surface will not be losing heat to the surroundings. The steam consumption will be less due to the higher latent heat at the lower pressure and to the reduction in loss. There will be less flash heat in the condensate.

Drying by hot air can be even more wasteful if the drier is not fully loaded. All the time the air is being heated, steam is being used. The amount of actual drying done by the hot air has no effect on the steam consumption. It is therefore most important that a hot air drier be kept fully loaded and then shut down.

749. MECHANICAL WATER REMOVAL. This is probably of greatest importance in textile processes, dyeing, bleaching, laundering, paper-making. The removal of gross quantities of water from such things as textiles can be done far more economically by mechanical means, that is by mangling, squeezing or hydro-extraction, than by heat, by calender, press, tumbler, tunnel or chamber.

In the laundry discussed in Chapter 20 the amount of dry laundry handled was estimated at 38,000 lb./week. If the hydro-extraction in this laundry could be improved from 50 per cent. to 45 per cent. of moisture in the goods leaving the hydro, there would be a saving of water to be evaporated of 1,900 lb./week which would save some 4,000 lb. of steam or some 7 cwt. of coal a week.

In the making of paper the opportunities of removing water mechanically rather than by steam heat are immense. The web as it enters the machine is really a web of water entangled in fibre. It leaves as a web of fibre almost dry. The water is removed mechanically before the web passes on to the steam heated drying cylinders. The removal of water from paper gives us another excellent example of the danger of thinking in percentages unless the meaning of the percentage is fully understood.

The dilute pulp or "stuff" is put on to the porous band at the wet end of the machine with about 99 per cent. of water. The suction and squeezing arrangements or couch rolls remove say 96 per cent. of the water which brings the composition of the web down to say 25 per cent. fibre and 75 per cent. water. This water must be removed by the steam-heated cylinders of the machine. Now suppose the suction and couching could bring the water content down to 70 per cent. instead of 75 per cent., what would be the saving ?

	25 parts of fibre and	75 parts of water
are dried to say	93 ,, ,, ,, ,,	7 ,, ,,
from	93 ,, ,, ,, ,,	279 ,, ,,
Evaporation		272 ,, ,,
	30 parts of fibre and	70 parts of water
are dried to	93 ,, ,, ,, ,,	7 ,, ,,
from	93 ,, ,, ,, ,,	217 ,, ,,
Evaporation		210 ,, ,,

By reducing the water content by 5 per cent. from 75 per cent. to 70 per cent. the evaporation has been cut from 272 to 210 or a reduction of 23 per cent.

Percentages can give very false impressions.

750. PROCESSING AT LOWER WATER CONTENT. The processing of water soluble materials is almost always easier at high dilution than at high concentration. If output is being pressed for or the process is giving trouble, the easy aid of extra dilution is very tempting. If the water has all to be evaporated the waste of steam is exactly proportional to the added water. Here is an example from the author's factory.

Sugar liquors can and should be filtered at 68 per cent. concentration. If the coagulation of the gummy impurities is badly done filtration is slowed and throughput suffers. The result is that to maintain the required throughput the liquor has to be diluted to say 64 per cent. With an output of 12,000 tons a week this results in the following :—

Instead of								
	68	tons of sugar in			32	tons of water,		
or	12,000	,,	,,	,,	5,647	,,	,,	,,
we have								
	64	,,	,,	,,	36	,,	,,	,,
or	12,000	,,	,,	,,	6,750	,,	,,	,,

Extra evaporation 6,750 — 5,647 = 1,103 tons of water per week, or
140 tons of coal per week.

751. EXOTHERMIC AND ENDOTHERMIC PROCESSES. Many chemical or physical changes give out or take in heat. Every effort should be made to turn these reactions to good economical use. Here is a small example that was operating in the author's factory 15 years ago.

The raw sugar is dissolved in water prior to filtration. The liquor must be filtered fairly hot, at about 170° F. A small portion is required quite cold—at nearly freezing point—for part of the coagulation process. This small portion had been taken from the bulk of the hot liquor and cooled by refrigeration. By withdrawing a small portion from the dissolving vessel in the form of undissolved crystal-water slurry, the heat of solution can be used for cooling. This saves the heat needed to provide the heat of solution the sensible heat of the solution at 170° F. and the power needed to refrigerate the liquor down to 40° F.

752. INCREASED YIELD. If the output of a batch process can be improved so that more finished material can be turned out of each batch, large savings in steam may be possible. Not only is there a big reduction in the fixed steam needed to make good the losses, but in some cases there is less material to be processed in the next stage.

When sugar is crystallised from a sugar solution standard technique produces about 45 per cent. of the sugar as crystals from each batch. The mother syrup is separated in a supersaturated condition and must be at once diluted to below saturation. All this diluting water must be evaporated in the second crystallising process. By improving the plant and technique, the crystal yield in the author's factory has been increased to about 63 per cent. This has saved 10 per cent. of the steam used in the factory, has reduced the plant by two-fifths and has produced a better quality product.

753. USING WASTE STEAM—FLASH STEAM. The collection of waste heat and its second or third use is probably by far the largest heat saver, but the matter must be approached with deliberation and circumspection. Many engineers when commencing a heat campaign start on the collection of flash steam and condensate. Now unless there is a use for the flash steam it is no use collecting it. Unless the hot condensate can be pumped there is no use returning it. Of course there is almost always a use for flash steam and a method of handling hot condensate, but these matters must be gone into before the attack starts.

If the works is composed of a spread of single-storey buildings the collection of flash steam and condensate will be relatively expensive, and the amount of heat recovered is always relatively small. It is very unlikely that flash steam or condensate are the most lucrative, large or likely losses to tackle.

Turn back to the laundry diagram, Fig. 355, in Section 605. The condensate is collected, the flash is lost. Let us write these down together with another item :—

Condensate collected	647	Btu/lb. dry laundry.
Flash lost	278	,, ,, ,,
Engine exhaust unaccounted for	2,818	,, ,, ,,

It would surely be madness to bother about flash collection here. There would be nothing whatever to use the heat for. The laundry management think that they are using all their engine exhaust but they are doing nothing of the kind. The laundry must be viewed as a whole, and if it is, as in Fig. 356, Section 610, we can use the flash steam all right after we have put the rest of the plant in order.

Similarly in Fig. 375 the flash steam loss is 169 Btu and the meal drier is taking 2,655. From Fig. 378 we see that this 2,655 can be replaced by waste heat made up with coal heat of 590. It is again silly to bother about flash loss until these major faults have been found and dealt with.

But all these matters are quite hidden until a heat balance or picture of some kind is constructed.

754. USING HEAT A SECOND TIME—VAPOUR COLLECTION. The brewery offers excellent examples of the uselessness of haphazard waste heat collection. Suppose a brewery is hot-sterilising its gravity adjusting water. A heat saving enthusiast comes along and points out the great saving—some 20,000 Btu/barrel—that will result from cold sterilising with ozone. What is the result ? There will be 20,000 Btu less to be recovered from the frig. and consequently these 20,000 Btu must be put into the mashing liquor otherhow—probably by live steam.

Again, suppose the enthusiast views the cloud of cask steamings with disfavour, and collects it in a spray condenser for heating the cask washing water. If the copper vapour had been used for water heating, the use of other vapour will simply mean that some copper vapour must be blown to waste.

All waste vapour should if possible be used, but it cannot be used unless a picture is made which will show all the various vapours and the possible uses to which they can be put. In the case of the brewery it would seem that all the waste heat can be used, but there is always the alternative of trying to avoid wasting heat in the first place. For example, it is better perhaps to wax the casks, and thus dispense with the need for steaming, than to try to find a way of using the waste steaming steam.

755. USING HEAT AGAIN—MULTIPLE EFFECT. If evaporation is being done and there is no possibility of using the heat in the vapour from the evaporator, then the conversion of the plant to a multiple effect is a primary heat saving and should be done forthwith. There are many evaporating jobs where multiple effect working is out of the question, or at any rate limited, on the material being concentrated. Milk, for instance, or fruit juices, cannot tolerate temperatures above certain maxima lest vitamins be destroyed. This greatly limits the possibility of multiple effecting. But it may be possible to combine the evaporation of several different materials in one multiple effect evaporator, where the evaporation of the more robust material is carried out in the earlier and hotter effects and the more delicate material in the later and cooler effects.

An evaporator working in single effect can often be converted to multiple effect for the production of pure distilled water for boiler feed or for process. If the material must be evaporated at very low temperature the water evaporating effect can be the first effect, because water does not mind how high the temperature is at which it is boiled.

The real key to efficient multiple effect working is the use of the later vapours for sensible heating, and this has all been explained in Chapter 17.

756. THE APPROACH TO STEAM SAVING. There is no cut and dried technique. Probably the best plan is to set in motion all the primary heat savings :—

Lagging steam pipes and process pipes
Stopping leaks
Reducing safety valve blows
Ensuring good trap maintenance and operation

While these things are being got under way a start can be made in taking out the balance and the establishment of bogey. Before attacking the collection of waste heat every effort should be made to do all that can be done in the way of reducing the work that the steam has to do by :—

Reducing process temperatures
Reducing reprocessing
Quick processing
Keeping plant fully loaded
Mechanical water removal
Processing at lower water content
Using Endo- or Exothermic reactions
Increasing yield
Reducing power load

All this will take some time and will give ample opportunity to get out a real picture of what is really going on in the factory and what might be done. These primary heat savings and processing improvements will probably open the eyes of the disbelievers, and will make the whole factory steam-conscious so that the co-operation of everyone will be available for putting the target plan into operation. A re-organisation of any magnitude will take several years. The savings that are possible may in some cases not be sufficient to warrant doing the work. But one day that part of the plant will need renewal and the plant can then be attacked piecemeal as opportunity offers.

757. THE FINANCIAL ASPECT. Managements are very queer. They will cheerfully appoint a man at £1,500 a year to push sales or to purchase materials, and will be quite content if the man repays his salary; more often than not he only slightly repays it and they are still pleased. They will gaily order a £4,000 machine if it saves a couple of girls. But they tend to be allergic to any proposal to spend anything that will give a permanent saving of steam.

Most modern laundry plant is expensive—a big washer with all the latest frills may cost about £4,000; a big ironer may cost about £4,000. Now all these things USE steam. A heat exchanger in the wash house drain will also perhaps cost £4,000, there is no saving in girls, there is no obvious increase in output but it does invisibly SAVE steam. Hardly any laundries have got such heat exchangers, yet the return is substantial though it is not by any means obvious. In the case of the laundry discussed in chapter 20, wash house drain heat recovery would save about 1¾ tons of coal a week, about 3,000 gallons of water and its softening and pumping—a pretty certain saving of nearly £12 a week. In addition there are the following hidden advantages: the boiler load is reduced by 10 per cent. This means that the laundry output can be increased by 10 per cent. before a new boiler need be considered. The existing pumps, the well, the steam and condensate pipes will all carry an additional 10 per cent. laundry throughput before being overloaded. Why has such a proposal so little appeal to management? Probably because the management does not really understand the matter and very naturally distrusts something it does not fully understand. One of the objects of this book is an endeavour to make these things more understandable to managements.

Management has usually suffered from the optimistic estimates of the engineer and the technical salesman. Almost every manager has at some time or another (often nearly every time) been told that a piece of plant will cost £1,000 and can be in operation in six months. It does in fact cost £1,500 and is not working in ten months!

Estimation is usually optimistic and the estimation of the cost and time of erection of steam saving plant will probably be no less optimistic than that of other plant, but there is one very rosy exception regarding optimistic estimation, namely the savings that will result. In the author's experience the amount of saving that a steam-saving job is estimated to yield is nearly always in fact exceeded. The reasons for this are to some extent clear. When calculating steam savings the direct coal saving is usually the only item taken. Now there are all kinds of consequential benefits. A saving in coal means a saving in coal handling and in ash handling; a reduced boiler load will bring a reduction in boiler maintenance; less water must be softened; less water

must be bought and pumped ; plant that was overloaded becomes correctly loaded, etc., etc. The consequential savings often ramify in all kinds of unsuspected directions. For example, a steam saving will result in a lightening of the boiler load so that it might be possible to burn a cheaper fuel. A boiler on three-quarter load is almost always more efficient than one on full load. So that it is fairly certain that steam savings will always eventually result in a greater money saving than can be estimated.

There is also another great benefit in steam savings that are obtained by alterations to plant and that is that they go on without supervision. Screwing up the boiler efficiency may be very profitable, but to maintain the high efficiency calls for unceasing vigilance and supervision. The use of waste heat, or the reduction of processing temperature requires the minimum of supervision, in fact many of the things discussed in this and in preceding Chapters require no supervision whatsoever.

We will now look at three examples of steam savings. As the heat balance has been dealt with in Chapter 20, the balances will not be given in these examples and the industries discussed will be different. It must be remembered that these three examples were war-time examples and the whole reorganisation was conditioned by what could be done, rather than what ought to have been done. The three following examples are all small works, not very good at thermal technique, but typical of many hundreds throughout the country. All showed a most laudable willingness to co-operate with the Ministry of Fuel engineers who made the various recommendations, but all suffered from conservative tradition that required some persuasion to overcome.

758. **FORGE.** This is a small shop with 14 half-ton steam hammers, which take 5,000 lb./hr. on average—see Fig. 412. The hammers exhausted into the top of the unlagged feed water tank, from which most of exhaust escaped to atmosphere. The space heating was done with live steam reduced to 10 psi and taking 3,000 lb./hr. in winter.

Two thirds of the space heating condensate was lost. The single Lancashire boiler that supplied the steam was severely pressed to produce the 9,000 lb./hr. required in winter and its combustion was poor and the efficiency low. The boiler feed pump exhausted to atmosphere.

Fig. 413 shows the arrangement after very simple reorganisation. The space heating is run at approximately atmospheric pressure and takes almost entirely hammer exhaust. The small occasional live steam make up is controlled by a thermostat. The space heating condensate is all recovered and being occasionally under vacuum, is pumped by an extraction pump to the feed tank which has been lagged. (As the hammer exhaust is very wet more steam is allowed for space heating than formerly.)

The surplus hammer exhaust is blown into the water in the feed tank and a coil in the tank takes the feed pump exhaust. To prevent trouble at the pump suction with the hotter feed, the feed tank has been raised 10 feet.

Oil separators and vacuum breakers were fitted into the two pipes leading exhaust and condensate back to the feed tank.

The direct steam saving was 3,700 lb. steam/hour or 41 per cent. in winter.

There were two indirect savings. As the boiler could now easily take the load its efficiency improved by 5 per cent., and it was possible to reduce the grate area. Due also to the lighter boiler load it was possible to use a cheaper coal, rough smalls instead of trebles.

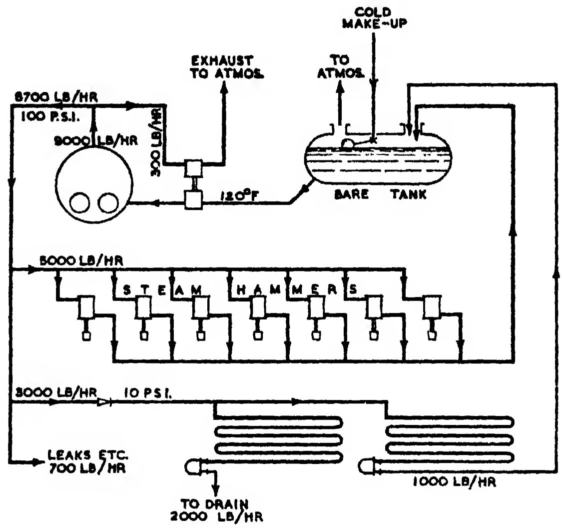


FIG. 412. SMALL FORGE BEFORE HEAT SAVING CAMPAIGN

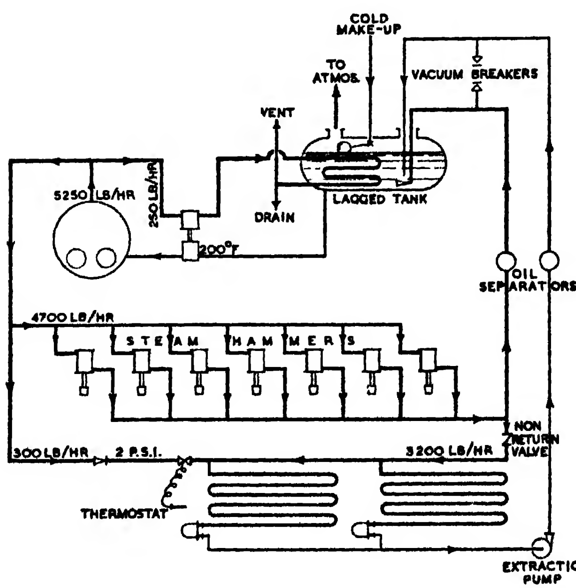


FIG. 413. SMALL FORGE AFTER HEAT SAVING CAMPAIGN

Here is a statement of the savings and costs in 1943 :—

	Cost	Yearly saving: Tons, coal	Yearly saving: Money
	£		£
Hammers overhauled. Pump exhaust to coil in feed tank. All condensate returned. Return pipes submerged. Vacuum breakers fitted, leaks stopped. Pipes lagged, etc., etc.	500	500	1,000
Hammer exhaust to space heating. Thermostat control. Extraction pump. Oil separators	1,350	200	400
Improvement in boiler efficiency	50	100	200
Cheaper coal—3*s*. per ton			say 150
	£1,900	800 tons	£1,750

The coal saving amounted to—

29 per cent. in summer, and
49 per cent. in winter.

759. **JAM FACTORY.** Fig. 414 shows the original arrangement where the main steam users are six hemispherical unlagged jacketed pans supplied from a vertical boiler burning 7½ tons coal/week in winter and 6¾ tons in summer.

The other steam users are the pulper, used intermittently, the jar washing plant, under inadequate control, and the low temperature space heating of the warehouse.

The pans were group-trapped, the traps discharged to drain, and the pan blow-off valves to atmosphere.

The reorganisation is shown in Fig. 415.

The boiler was relagged and a damper fitted to the stack.

The pans were lagged and each was fitted with its own trap and air vent.

The space heating steam pressure was reduced to 15 psi by a reducing valve and was fitted with thermostatic control.

All condensate was returned to the feed tank or to the jar washer.

The steam pipe to the pans was drained and trapped, its condensate going to the jar-washing tank into which also the pan blow-off was piped.

As the water temperature in the feed tank was now too hot to be handled by the injector, a feed pump was installed and its exhaust used to add still more heat to the feed.

In order to ensure that all the feed pump exhaust heat is used, a thermostat is fitted to the feed tank so that when its temperature rises too high the condensate is diverted to the jar-washing tank.

In order to prevent "gushing" into the jar tank an orifice is fitted in the pipe bringing condensate and blow-off steam to the jar tank. A stand pipe is fitted to this line so that when a "gush" occurs it is throttled by the orifice and cushioned by the stand pipe.

The steam supply to the jar-washing tank is thermostatically controlled.

The process performance was greatly improved.

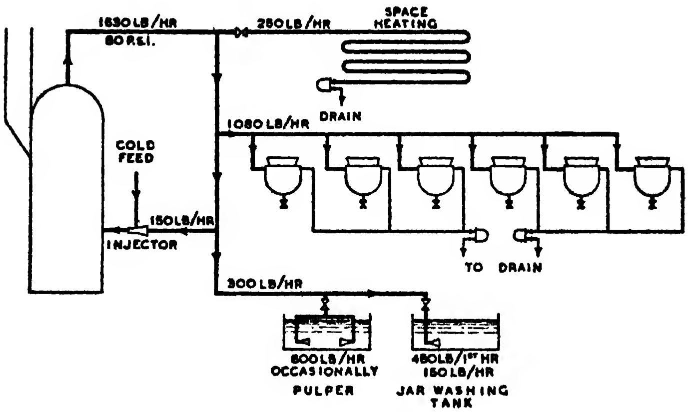


FIG. 414. SMALL JAM FACTORY BEFORE HEAT SAVING CAMPAIGN

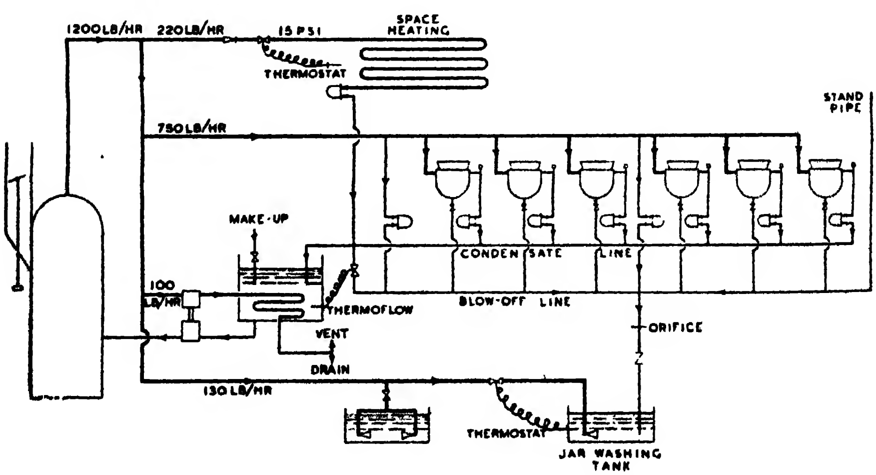


FIG. 415. SMALL JAM FACTORY AFTER HEAT SAVING CAMPAIGN

Here is a statement of the savings and costs in 1944 :—

	Cost £	Yearly saving Tons, coal	Yearly saving £
Thermoflow to feed tank. Thermostatic control to jar-washing tank. Blow-off piped to wash tank, orifice and stand pipe	75	19	38
Lagging pans	150	95	190
Separate traps to pans. Condensate return. Lagging boiler	230	66	132
Thermostatic space heating (winter only)	30	4	8
	£485	184 tons	£368

760. PAPER MILL. A very old small plant, working 136 hours a week, shown in Fig. 416.

The paper machine is driven by two engines, almost the whole of the exhaust from the smaller engine going to atmosphere.

The drying cylinders were group-trapped and had no condensate scoop. The super calender condensate was wasted.

The digesters were unlagged and leaking.

The triple effect evaporator was unlagged.

Digesters and evaporator took exhaust steam from the back pressure electric generating plant.

The space heating and the size pot were fed with live steam reduced to 80 psi.

The coal consumption was 116 tons per week.

The reorganisation is shown in Fig. 417.

The smaller paper machine engine is replaced by a motor.

The drying machine cylinders are fitted with individual traps and condensate scoops. This resulted in a reduced load on the cylinder drive from 100 H.P. to 92 H.P. due to the reduction in condensate that was being churned inside the cylinders.

There was now insufficient exhaust steam from the one engine to supply the paper machine, so a make-up valve reducing from 180 psi to 25 psi was fitted.

The condensate from the evaporator was sent to the digesters.

The space heating was reduced to 5 psi and taken partly from paper machine flash and partly from engine exhaust.

The digesters and evaporator were lagged and the leaks stopped.

The exhaust from the boiler feed pump was used to heat the feed.

The size pot was put on to exhaust steam and lagged.

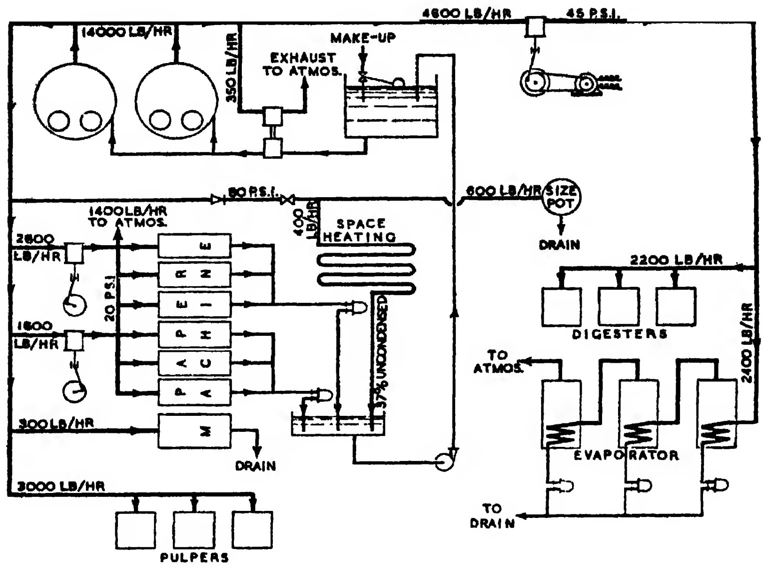


FIG. 416. SMALL PAPER MILL BEFORE HEAT SAVING CAMPAIGN

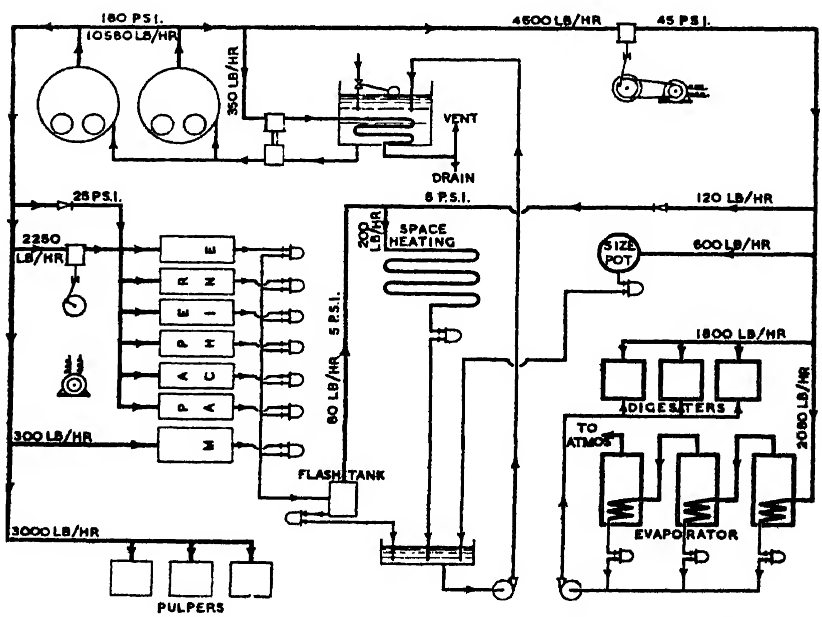
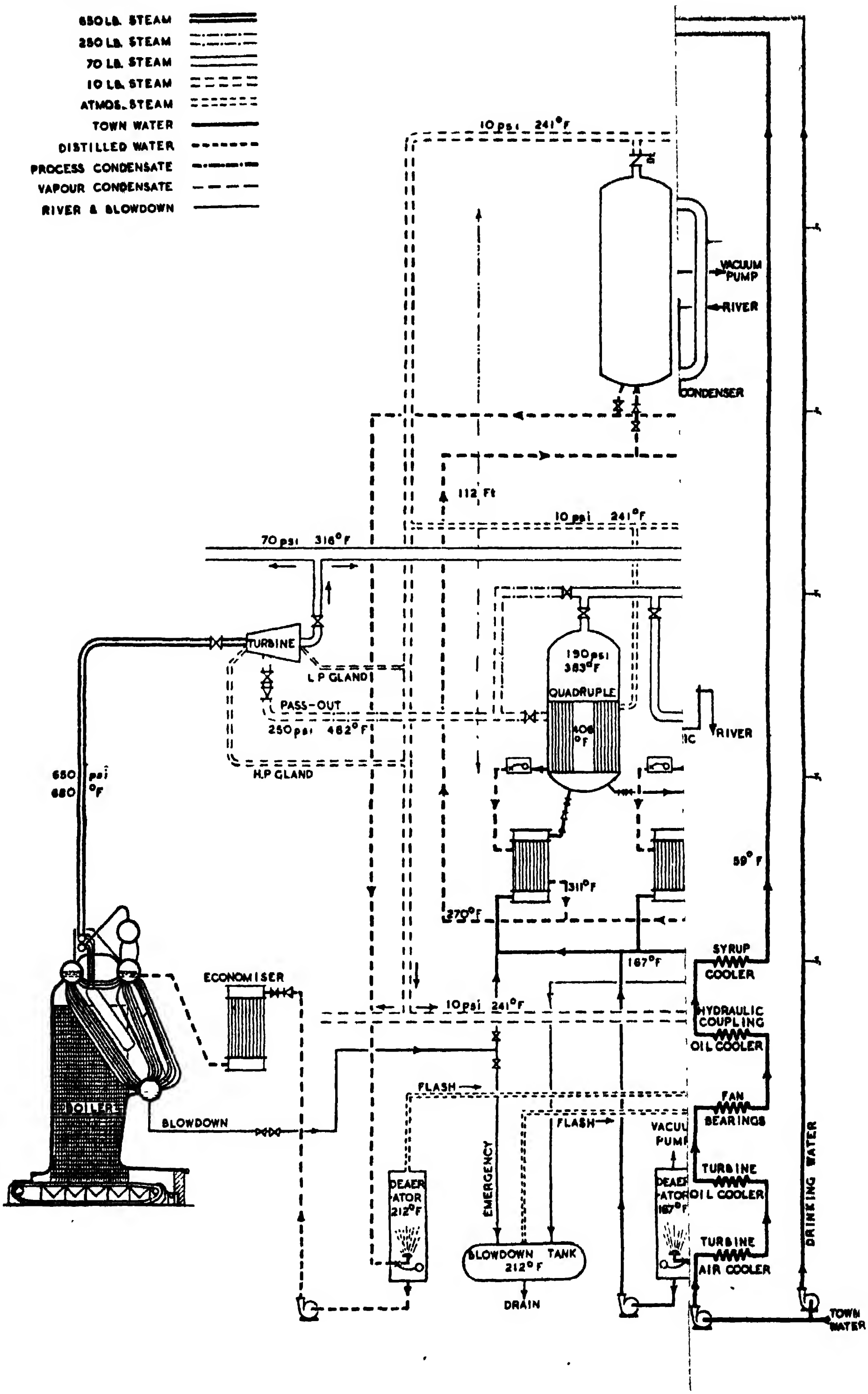


FIG. 417. SMALL PAPER MILL AFTER HEAT SAVING CAMPAIGN

[To face page 771]



Here are the savings and costs in 1944 :—

	Cost	Yearly saving	
	£	Coal	£
Motor for paper machine. Make-up valve for paper machine steam. Individual traps and water scoops	350	630	1,260
Less cost of bought current, 30 kW at 1*d*.	—	—	850
Less coal used by grid	—	137	—
	350	493	410
Lagging digesters and evaporator. Lagging size pot and using exhaust steam. Using feed pump exhaust. Collecting paper machine flash and reducing space heating to 5 psi. Reducing leaks, etc.	480	690	1,380
	£830	1,183 tons	£1,790

761. AN EXAMPLE OF HEAT SAVING TECHNIQUE. The following is a description of the steam and water circuits in the author's factory. The reorganisation was done over a period of about four years. The savings were given in Section 559 and amounted to nearly 50,000 tons of coal a year, which is a good enough excuse for including a description of the arrangements here.

The steam and water circuits are shown in Fig. 418 and the arrangements include the following devices :—

Back pressure pass-out turbine.
Collection and use of turbine gland steam.
Continuous blowdown.
Multiple effect recovery of blowdown heat.
Multiple effect evaporation.
Collection of flash steam.
Flash evaporation for cooling liquors.
De-aerators.
Spray vapour condenser for water heating.
Steam accumulator.
Hydrostatic legs to trap for widely different pressures.
Condensate handling :—
    By pump.
    By barometric leg.
    By steam pressure.
Extensive space heating by waste vapour.
Extensive hot water heating by waste vapour in cascade.
Heat exchangers in cascade.
Thermostatic selection with overriding flow control.
Evaporator replacing reducing valve.

(*a*) *Town Water.* At the bottom right hand corner of Fig. 418 the cold water enters the refinery. It first reaches the drinking water pump, which pumps a constant quantity continuously through the refinery drinking water loops. This quantity is less than the minimum factory use so that the flow through the drinking water line need never be reduced.

The remaining cold water passes through all the low temperature coolers on its way to the storage tank, gaining about 9° F. in the process. From the storage tank there are four outlets : hydrants used only in case of fire ; social services (lavatories and the cold supply to basins and baths) ; cold water to process ; and lastly the main outlet to process and boiler feed. The social services always call for more than the drinking water pump can supply. The cold water to process is small in average but takes place in large widely-spaced peaks. The great bulk of the cold water leaves the storage tank by the left-hand outlet into the circulating tank.

(*b*) *Town Water Heating.* From the circulating tank the water passes to the right through the surface condensers attached to the vacuum pans which work on batches on a 1½ to 3 hour cycle, and whose vacuum varies between 20 in. and 27 in. In order to get the best possible heat recovery a controller is fitted to these condensers. This controller has been described in detail in Section 537. Each thermostat T_1, etc., opens the valve V_1, etc., for a rise of temperature on the condenser to whose outlet it is attached. The flow control in the metering tank sees to it that the valves are opened sufficiently to pass the water demanded, while the thermostats see to it that the most water passes through the hottest condenser. In this way all the town water is heated to about 135° F. by vapour whose heat would otherwise have gone to the river through the jet condensers.

(*c*) *Factory Heating and Domestic Hot Water.* The water flows from the condensers through all the factory heating loops and supplies the basins, baths, showers and sinks. It returns to the circulating tank helped on its way by a booster pump. It then flows with any new cold make up water back through the condensers for reheat.

Owing to its comparatively low temperature it is necessary to augment convection by a large capacity booster pump to secure the necessary heat transfer in the space and air heaters. Heat is thus supplied to some 15,000 sq. ft. of radiator surface, for heating some 200,000 cu. ft. of air per minute, and hot water for some scores of basins, some dozens of baths and showers and for sundry sinks. The heat recovered in this waste heating system represents about 8 per cent. of the coal consumed on the boilers.

(*d*) *Town Water Heating.* After circulating a number of times through these heating loops the water at about 135° F. passes through the tubes of a surface condenser attached to a sugar liquor evaporator. The primary function of this evaporator is to cool the liquor from filtration temperature to the temperature suitable for use in the next process. This is done by flash under the appropriate vacuum. The evaporator is, however, also fitted with a calandria supplied with low pressure steam. The steam put into this calandria replaces the steam formerly blown direct into the reaction tank of the water softening plant and replaces some of the steam used in the Hawley boiler for process water. By transferring the heat to the water by means of vapour from the sugar solution,

the water is heated by the same amount as previously and the liquor is concentrated by between 2 and 3 per cent. for nothing. (The value of 1 per cent. concentration is about 15 tons of coal a week.)

The water, now heated to about 165° F., then passes through a heat exchanger where it removes some of the excess heat from the process condensate. The town water now has a temperature of about 180° F. and it flows either to process or to the water softening plant for the boiler feed. If it goes to process it passes through the Hawley boilers which were described in Section 383.

(*e*) *Temporary Hardness.* It will be seen that the unsoftened raw water is heated to about 180° F. and it might be thought that there would be a heavy deposition of temporary hardness scale on the heat exchangers and condenser tubes. This does not occur because the hotter the water is, the higher is its pressure, since, by the accident of plant arrangement, the hot plant is lower in level than the cooler plant. There is however a very large aeration of the water in the circulating tank and there is some corrosion of the space heating and air heating plant.

(*f*) *Softening Plant.* There is a heat loss of about 10° F. in the water softening plant. This is replaced by the water spraying through an extension to the surge tank which forms a spray condenser which recovers the heat in all the flash vapour that is piped to it at atmospheric pressure.

(*g*) *Distillation Plant.* The hot softened water is pumped into the quadruple effect still, which takes steam passed out from the first cell of the turbine at about 250 psi, and supplies the surplus steam that is called for by the process over and above that provided by the turbine exhaust. The still thus acts as a reducing valve between the turbine pass-out and the process steam main. The plant is normally worked in quadruple effect, but if there is a large process steam demand and the supply of distilled water is adequate, the plant can be worked as a triple or a double. It is so valved that any of the four bodies can be isolated. The softened water passes into each body through a heat exchanger which removes much of the excess heat from the distillate—or condensate. Each vessel is fitted with a feed water regulator of a different type, no regulator having as yet been found which is satisfactory for this plant. The water level is about one-third up the calandria tubes and no feed regulator seems to be able to know what the level really is.

The continuous blowdown from the first body goes into the second body where it parts with its flash which then works in triple effect. The blowdown from the second body carrying the first body blowdown goes into the third body where the flash then goes to help evaporation in the fourth body. The third body blows down into the fourth body and the flash joins the steam from the fourth body which goes into the process main. The continuous blowdown from the fourth body goes into the blowdown tank where its flash is piped to the centipede—the spray condenser attached to the water softener surge tank. The town water, as such, has now ceased to exist.

(*h*) *Distillate Handling.* The distillate, or condensate, from the still calandrias passes through float traps having counterpoised floats of solid aluminium, described in Section 292 and shown in Fig. 131. The condensate, or distillate,

is lifted by the pressure in the calandrias up to the distilled water storage tanks. As these carry safety valves set at 15 psi.g. and as their safe high level is 112 feet above the calandria bottoms there must be a minimum pressure in the last calandria of 112 ft + 15 psi + margin = say 70 psi.g. As the average temperature of the distilled water is often more than the temperature corresponding to 15 psi the excess heat flashes through the tank safety valves into the 10 psi process main.

The distilled water passes to the boiler feed pumps through the deaerator where it flashes down from about 240° F. to 215° F., the flash steam being piped to the centipede. The deaerator is shown in Fig. 418 at feed pump level. This is only for diagrammatic clarity ; the deaerator is actually some 120 feet above the feed pump to ensure a good suction head and to reduce the work done by the pump.

(*i*) *Boilers and Turbine.* The boilers generate steam at 650 psi.g. superheated to about 680° F. The continuous blowdown from the boilers goes into the first body of the still where its flash is used in triple effect.

The 650 psi steam goes to the turbine where about three-quarters of it passes right through to exhaust to the process main at 70 psi.g. The leaks from the two glands are piped to the 10 psi process main.

At the back of the first wheel about one quarter of the steam is passed out at about 250 psi to the still where its latent heat is used four times, before going to process where much of its heat is used twice more.

The plant was designed as a straight back pressure plant with steam supply to the still reduced direct from the 650 psi main. Steam savings were so great however as to upset the whole steam/power ratio. By good fortune a 5 in. branch was found in the right place on the turbine casing so that, after a little blanking off of diaphragm nozzles, the machine could work fairly well as a pass-out machine.

(*j*) *Vacuum Pans.* The main users of 70 psi steam are the white sugar vacuum pans, which boil under a vacuum which varies according to the state of the batch from about 20 in. to about 27 in. The vapour from these pans passes through the surface condensers which do the initial heating of the town water and provide the factory heating. There is always an excess of pan vapour which is condensed in the jet condensers and whose heat is lost.

(*k*) *Condensed Vapour Handling.* The condensed vapour from the surface condensers is extracted by means of barometric legs and atmospheric tanks. This condensate being sweet is passed to the process for dissolving crystals and for syrup dilution. Part of this vapour condensate goes through sparge pipes fitted in the pan save-alls to keep them washed free of sugar. To prevent uneven washing when the amount of condensate is small, the sparge pipe is arranged to take a quantity which is substantially less than the average flow. The remaining condensate overflows direct down the barometric leg.

(*l*) *Process Condensate Handling.* The condensate from the vacuum pan calandrias goes into a large common trap tank 130 feet below. Each pan has its own tail pipe and the capacity of the trap tank is equal to the combined capacities of the tail pipes. This arrangement enables one trap to deal with all the pans in one building and consequently the trap can be something better

than usual. This trap is described in Section 292 and shown in Fig. 130. Owing to the length of the tail pipes it is possible for one calandria to carry 70 psi while its neighbour is working at 16 psi. The 70 psi tail pipe will be empty but the 70 psi pressure will be balanced in the other tail pipe by its pressure of 16 psi and the hydrostatic head of its full 130 feet of water. From the top of the trap tank a small permanent leak is piped into the 10 psi main to vent the air.

From the trap tank the condensate goes into the flash tank whose vent is connected to the 10 psi main. The condensate is then pumped to storage through a heat exchanger where it parts with most of its excess heat to the now hot town water.

At times owing to small process demand for steam, there is insufficient steam passing through the still to provide enough distilled water for boiler feed. This deficiency must be made up from process condensate, which is liable to be contaminated with sugar. At high temperatures sugar is broken down into organic acids and sweet feed water for high pressure boilers is therefore not permissible. The process condensate pipe passes the laboratory where a continuous alpha-naphthol sweet test is done. Whenever the condensate is sweet-free and there is a shortage of distilled water, the laboratory closes the valve P admitting the process condensate to its storage tank. It then forces its way into the distilled water tank through the non-return valve.

(*m*) *Ruths Accumulator*. Any surplus steam in the 70 psi main passes into the accumulator whence the 10 psi main can draw when the various flashes and gland leaks are insufficient.

Cross-connection, by-passes, surplus and reducing valves for peaks and emergencies are omitted from the diagram for simplicity.

762. 100 PER CENT. DISTILLED MAKE-UP. In the foregoing description of the steam and water arrangements in the author's factory the provision of boiler feed constitutes a major process with a large and costly plant. The matter merits consideration because there are several ways of going about it. The problem arises in many factories, in fact in all those where it is necessary to raise steam at very high pressure to generate the needed power and where the process condensate may become contaminated and unusable in high pressure boilers. Other factories are those, such as explosives factories, where the works are so spread out that the return of condensate is out of the question due to capital cost.

Fig. 419 shows the original arrangement in the author's factory, with very round figures. Three quarters of the boiler steam passed through the turbine which produced 1 kWh per 30 lb. of steam. The remaining quarter of boiler steam was reduced to 250 psi and put into a quadruple effect still exhausting to process at 70 psi. The overall result was that 40 lb. of steam was passed to process for each 1 kWh generated.

Steam savings in the process were such that it was impossible to spare the steam needed for the evaporator. A suitable pass-out point fortunately was available on the turbine casing and the arrangement shown in Fig. 420 was adopted. This was a marked improvement.

If the plant were being designed afresh, it is possible that the arrangement shown in Fig. 421 might be adopted. The whole of the steam is passed through the turbine, whose back pressure is raised 10 or 15 psi. The exhaust then goes

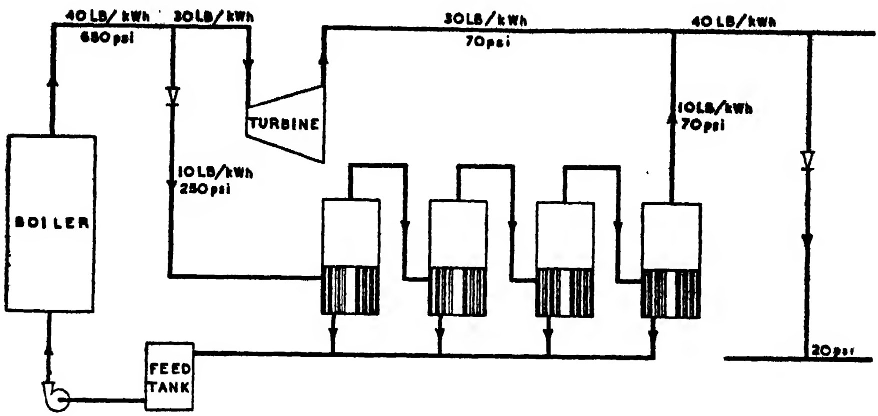


FIG. 419. BACK PRESSURE TURBINE AND QUADRUPLE EFFECT FEED WATER STILL FED WITH LIVE STEAM

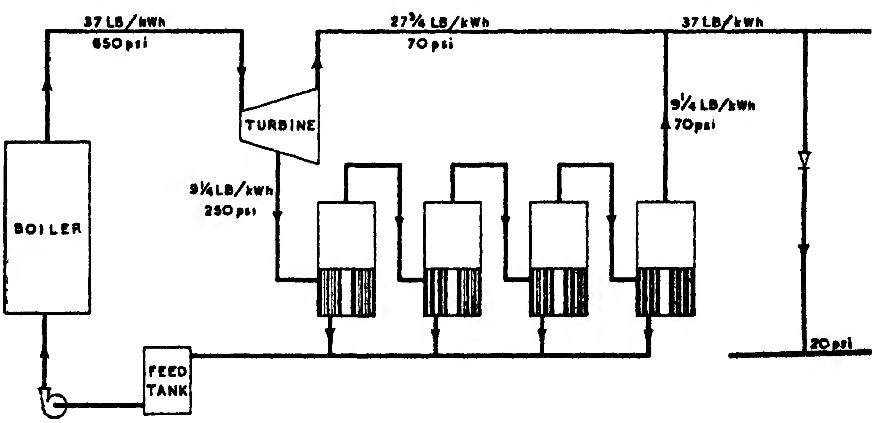


FIG. 420. PASS-OUT TURBINE FEEDING QUADRUPLE EFFECT STILL

into a single effect low-pressure evaporator which exhausts into the 70 psi main. The higher back pressure lowers the heat drop from 123 Btu to 110 Btu, so that the steam consumption will be about 33½ lb./kWh. This is a substantial saving and requires only a low pressure evaporator.

Fig. 421 shows the reducing valve from the 70 psi main to the 20 psi main replaced by a small evaporator which will give a margin of distilled water to make up for loss due to blow-down and soot-blowing.

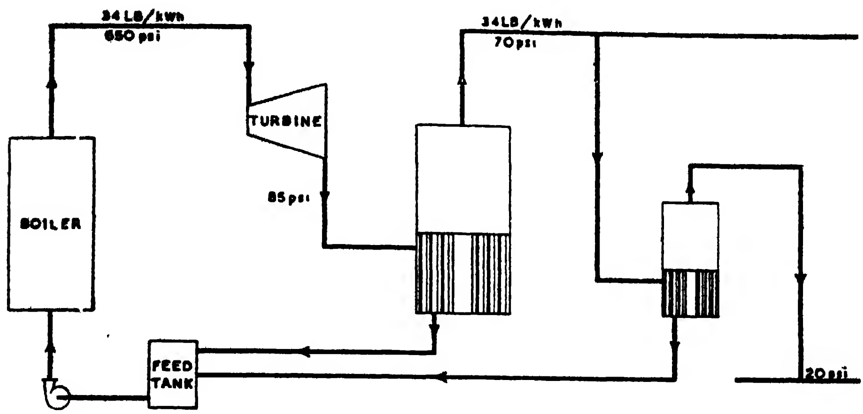


FIG. 421. BACK PRESSURE TURBINE EXHAUSTING INTO SINGLE EFFECT STILL WHICH FEEDS PROCESS. LOW PRESSURE REDUCING VALVE REPLACED BY STILL

CHAPTER 25

STEPPING UP HEAT

> **Heat cannot of itself pass from a colder to a hotter body.**
> R. J. E. CLAUSIUS. *Second Law of Thermodynamics*. 1850.

WHEN heat has become so degraded that it is useless for heating purposes, it generally has to be thrown away. Sometimes the cost of throwing it away is considerable, for example in cooling towers. There are sometimes circumstances when it pays to upgrade this heat by raising its temperature or potential. The increasing cost of coal makes the possible limits within which upgrading is economical much wider.

763. THE VIRTUE OF HEAT. The second law of thermodynamics has been stated in all kinds of ways, many of them incomprehensible to ordinary folk, but Clausius has set it out in simple words that any one can understand. We accept this law, especially in this form, because it states a fact that is a matter of everyday observation and experience, although its consequences are not always clear.

We are inclined to think that one Btu is the same as any other Btu. This is not the case at all. Suppose we have 1 lb. of steam at atmospheric pressure, it will contain 1,151 Btu. If we pass this steam through a turbine exhausting into a condenser at 29 in. vacuum at a temperature of about 80° F., we have available 202 Btu for converting into power. The remaining 949 Btu, over 80 per cent., are useless and must be thrown away in the condenser at 80° F.

Now suppose we put our 1 lb. of atmospheric pressure steam into an air heater and heat air up to 70° F. We can use the whole of the latent heat and $212 - 80 = 132$ Btu of sensible heat before our 1 lb. is cooled to 80° F. So that we have used $971 + 132 = 1{,}103$ Btu out of the original 1,151, and we have to throw away 48 Btu, only 4 per cent., in the condensate at 80° F.

So that we see that there are different proportions of useful to useless heat units dependent on what we propose to do with our heat. We can compare them thus :—

	Power	*Heating*
Initial temperature	212° F.	212° F.
Final temperature	80°	80°
Useful Btu	202 Btu	1,103 Btu
Useless Btu	949 „	48 „
Total heat in 1 lb. steam	1,151 Btu	1,151 Btu

Now suppose we had 1 lb. of steam at atmospheric pressure with which we wished to do the evaporation of a very dilute aqueous solution under atmospheric pressure. The process would be impossible because there would be no heat potential or temperature difference to cause a flow of heat. For this purpose all the 1,151 Btu would be useless and unless we had some other process available to use heat at this potential it would all have to be rejected.

If we raised the pressure on the steam very slightly so that it had a temperature of 213° F. it would contain practically the same total heat and there would be 970 Btu of latent heat and 181 Btu of sensible heat. If we were in no hurry our steam at 213° F. would now transfer its latent heat to the liquid at 212° F. and we could carry out evaporation and could use all the 970 Btu of latent heat.

By the addition of less than 1 Btu we have made 970 out of 1,151 Btu available instead of none.

M. A. Thring has tried to devise a function wherewith to measure the usefulness or availability of heat, and he calls this function "virtue". The name and the idea are excellent, but the function is so abstruse as to make it of doubtful practical value.

Oscar Faber has suggested the classification of Btu into useful power-producing units for which he proposes the name "Carnots" and useless units which he proposes to call "Basic Units". This suggestion does not get over the difficulty that if we are going to use the steam for heating there are 1,103 available useful units whereas if we are going to use it for power there are only 202 available units.

We have seen that by raising the potential of heat by a very small amount a great deal of heat may be rendered available. It clearly behoves us to see whether any of the ways of stepping-up or up-grading heat can be economically done. As far as the author knows there are only three ways :—

(*a*) Thermo-Compression : 1. By steam injector.
2. By mechanical compressor.

(*b*) Heat Pump.

(*c*) Boiling Point Elevation.

Thermo-compression has already been discussed in Section 493, and there is no need to repeat what has been said. The heat pump and the B.P.E. cycle are not so well known ; both of them may have a future and they will be discussed in some detail.

764. THE HEAT PUMP. In 1852 a young man of 28 who had been professor of natural philosophy at Glasgow University for six years, suggested that a reversed heat engine could act as a "warming engine". His suggestion appears in nearly every book on heat engines or thermodynamics, where it is almost always dismissed as a curiosity. Although Lord Kelvin lived to the good old age of 83 he did not live long enough to see his warming engine put to practical use. Coal was too cheap. With coal at 1957 price, which may well be exceeded for many years, Kelvin's warming engine, or "heat pump" as it is more usually now called, merits serious consideration because, in favourable circumstances, there is no doubt that it would pay.

The subject has received occasional attention from time to time and some examples of heat pump have been built and worked, one at least on an ambitious scale. In 1930, Haldane read a paper to the Electrical Engineers on the use of the heat pump, primarily from the point of view of making electric heating economical. He described a heat pump that he had built and used, and,

while his results were quite striking, his temperature and pressure range were clearly unsuitable and the machine's performance could have been improved. There are several heat pumps in operation in the U.S.A. and some large installations in Zürich.

In order to understand the fundamental principle underlying the successful application of the heat pump it is necessary to compare the cycles of heat engines, refrigerators and heat pumps.

765. HEAT ENGINE CYCLE. Fig. 422 is the temperature entropy diagram of a heat engine or a refrigerator or a heat pump. We will first consider it as a heat engine although it is drawn for ammonia, which we should be unlikely to use as a heat engine fluid. The fluid used makes no difference to the general principles.

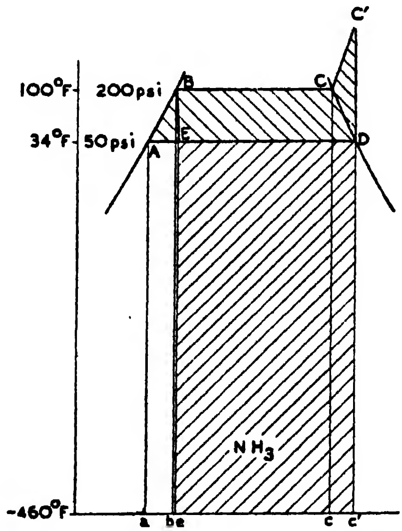


FIG. 422. HEAT ENGINE, REFRIGERATOR AND HEAT PUMP CYCLE

The liquid at A in Fig. 422 is heated to B the liquid boiling point appropriate to the pressure. This calls for the addition of the sensible heat area A B b a. From B to C the liquid is vapourised at constant temperature with the absorption of latent heat equal to area B C c b. Superheat, area C C′ c′ c is added and the final state of the vapour leaving the boiler to enter the engine is at C′. Adiabatic expansion takes place from C′ to D and the heat in the exhaust, area A D c′ a, is rejected along line D A.

The efficiency of this cycle is the work available, area A B C C′ D, divided by the total input heat area A B C C′ c′ a. In this case 11·8 per cent. It is clear from this picture, and the matter has been discussed at length in Chapter 2, that an increase in efficiency in a heat engine can be obtained by raising the initial temperature as much as possible and lowering the exhaust temperature as much as possible.

766. REFRIGERATOR CYCLE. Now let us look at the diagram from the refrigeration point of view. Cold low pressure vapour at D is compressed adiabatically to C′ where it is superheated by compression. From C′ the hot compressed gas is cooled to C by removing the superheat. All the superheat has been put in by the compressor and the amount, triangle D C′ C, of mechanical work is thrown away right from the start. From C the gas is cooled to B removing the latent heat and causing complete condensation. This cooling results in the rejection of the heat equivalent to area B C C′ c′ b. At B the cool liquid at high pressure is passed through a reducing valve to a lower pressure and as no heat is taken in or given up during this pressure-reducing process the state point will travel along the constant total heat line from B to E. As the total heat at B of the high pressure liquid must be the same as the total heat in the fluid after pressure reduction, and as the total heat in liquid at the lower pressure A is less than at B the fluid after pressure reduction to E will be partly vapourised by flash. The result is that at E there will be 90 per cent. liquid and 10 per cent. vapour.

The heat triangle B A E is equal to the narrow strip of heat E e b and is the forfeit we have to pay in this wiredrawing expansion process.

The cool liquid-vapour mixture is passed into the evaporator which absorbs heat from the space that is to be cooled and this heat provides the necessary latent heat to evaporate the liquid from E to D. The amount of latent heat absorbed from the cold room is area E D c′ e. The amount of refrigeration done is the amount of heat taken from the cold room, so that the efficiency of the refrigerator will be found by dividing the amount of refrigeration by the power put into the compressor :—

$$100\,\frac{\mathrm{E\,D\,c'\,e}}{\mathrm{A\,B\,C\,C'\,D}} = 638 \text{ per cent.}$$

767. HEAT PUMP CYCLE. The heat pump cycle is exactly the same as the refrigerator cycle except that we are concerned with the heat delivered at the hot end instead of the heat taken in at the cold end.

Cool liquid-vapour mixture at point E is passed through an evaporator coil situated where it can take up low grade heat from some more or less unlimited source, the air or a river. This source of heat causes evaporation of the liquid and supplies the latent heat, area E D c′ e. The cool vapourised fluid is then compressed by the pump to C′. We now have a fluid at about 180° F. instead of 34° F. containing heat equivalent to area E D c′ e, the heat taken from the river or air, and area A B C C′ D, the energy supplied by the pump. We cool the fluid by means of the air or whatnot that we wish to heat and the fluid gives up its superheat and then its latent heat area B C C′ c′ b. The high pressure liquid is then blown through a reducing valve from B to E and the cycle repeats. During the wiredrawing we see that we lose the narrow heat strip E e b as far as being able to absorb heat from the river is concerned. But we have used this strip of heat in our air heating process and it is equal to the work triangle A B E. As far as a refrigerator is concerned the triangles C C′ D and A B E are dead losses. As far as the heat pump is concerned they are

not lost but they represent so much extra power that has to be put into the pump and if they could be reduced or eliminated they would greatly increase the efficiency of the cycle.

The efficiency of the heat pump cycle shown in Fig. 422 will be the total heat pumped divided by the total heat energy supplied to the pump, or

$$100\,\frac{\mathrm{A\,B\,C\,C'\,c'\,e\,E}}{\mathrm{A\,B\,C\,C'\,D}} = 738 \text{ per cent.}$$

768. EFFICIENCIES OF OVER 100 PER CENT. Now it is customary amongst engineers to say that an efficiency of over 100 per cent. is impossible. So when a process crops up in engineering that has an efficiency of over 100 per cent. engineers divide the efficiency by 100 and call the result "coefficient of performance" or C.O.P. This is just pure cowardice. When the Almighty gives us a process where we get something for nothing, and whereby we can turn useless Btu into useful Btu why should we not welcome the process with open arms and boldly proclaim a three figure efficiency, instead of shamefacedly hiding it up and calling it C.O.P. ? The refrigerator or heat pump are not the only machines with efficiencies of over 100 per cent. We have seen in Chapter 17 that the multiple effect evaporator can also have an efficiency of several hundred per cent. The purists say that it is impossible to get something for nothing. This is not the case. There are several processes where something can be obtained for nothing as far as we mortals are concerned. If one sits in the sun the process of warming clearly has an efficiency approaching infinity. The mining of coal is a process having a thermal efficiency of many hundred per cent. The damming of a river to give water power is a similar process. We can now begin to see what it is that constitutes a process of over 100 per cent. efficiency. It is the collection and making available of God's gifts. The damming of a river, the getting of coal, sitting in the sun, the heat pump are all processes where we make heat provided by the sun more available. It is only when we come to use this heat energy that we fall down so badly.

769. HEAT PUMP TEMPERATURE RANGE. Clearly the smaller the difference of temperature over which a refrigerator or a heat pump works the smaller need be the power put into the compressor. So we must keep the top temperature T_1 as low as possible and the bottom temperature T_2 as high as possible. This is exactly the opposite of what we do when devising a heat engine cycle. If the temperature difference $T_1 - T_2$ were very small the efficiency of a heat engine would be negligible but that of a heat pump would be thousands per cent.

The hot and cold temperature limits must be looked at as must the level of the temperature range over which the cycle is to work. The object of a heat pump is to turn useless heat into useful heat. Now useful heat is heat with which we can heat a room, or with which we can generate power. We can heat a room easily with water at about 100° F. if we are careful of the design of heating plant, but to generate power we need steam at a temperature of say 200° F. to make the plant worth while. Let us take the case of room heating. There are some very modern buildings where the heating pipes are embedded in the floors. Such rooms are said to be beautifully heated—no cold feet and hot heads—ideal for directors' board rooms or trade union conference rooms,

or even the Mother of Parliaments. In order to prevent discomfort to the feet the floors should not be very much hotter than 75° F. To heat the floors to this temperature can obviously be done with sufficient circulation by water or other liquid at 100° F. If a building is to be heated principally by its ventilating air, this air must not be uncomfortably hot and probably must never exceed 75° F. Such air can be heated by a fluid at 100° F. So we can take 100° F. as being the lowest temperature down to which we can bring the hot end of our cycle.

What of the cold end ? We want to collect useless heat and turn it into useful heat. There is an almost unlimited supply of useless heat in most rivers, but in winter the river may get very cold and in winter we want to pump the maximum of heat. We must therefore fix the cold end as cold as is practical, bearing in mind that it must be as hot as possible from the point of view of thermodynamical efficiency. This apparent contradiction can be expressed another way. For efficiency we want to use the highest possible temperature, but this temperature must be well below the source of heat during the coldest period of the year. If the useless heat that is to be collected is in the river or in the air we must see to it that the evaporator pipes that are to pick up this heat never get below 33° F. lest they get coated with an insulating layer of ice. So that the low temperature end is fixed at about 34° F.

Fig. 422 is drawn for these temperatures—100° F. and 34° F. with ammonia as the working fluid. It is possible that other substances would be more suitable for a heat pump, but ammonia has certain advantages. Over the temperature range that we want the pressures are reasonable, and all its habits and properties are well known.

We see that from a purely technical point of view the heat pump should be worthy of consideration.

770. APPLICATION OF HEAT PUMP. Let us apply the theory to the case of a factory that is using 100,000 lb. steam/hour at atmospheric pressure to do the heating of its extensive buildings. Instead of raising low pressure steam let us raise steam at high pressure, using this steam to drive the heat pump ; the exhaust steam from the engine driving the heat pump will be exhausted at such a pressure that it can help the heat pump with the heating load. If we say that 100° F. is hot enough to do the required floor and air heating and if we allow a 13° temperature drop across the condenser, the machine driving the heat pump can exhaust at 27-in. vacuum.

Let us suppose that we use a turbine to drive the heat pump (we will skate over the difficulty of a flexible drive between turbine and pump), and let us postulate that we want to develop 100 H.P. per 1,000 lb. of steam. This will call for an actual heat drop of 255 Btu/lb. If we assume an efficiency ratio of only 60 per cent. the adiabatic heat drop will needs be 425 Btu. To get saturated exhaust at 27 in. will call for steam at 300 psi at 700° F. So that from each 1,000 lb. of steam we shall get 100 H.P. plus 1,030,000 Btu of latent heat in the exhaust.

Now 1 H.P. is equivalent to 2,545 Btu per hour so that we can tabulate the amount of heat that our combined turbine and heat pump will produce under various conditions of heat pump efficiency, and these are shown in Table LXXV. Apart from initial cost and upkeep the heat pump shows a big saving.

It will be seen that by the installation of a 40 per cent. heat pump the steam required is almost halved but that the coal consumption is only reduced by 42 per cent. The reason for this is that the steam needed to drive the heat pump is at a much higher pressure and contains much more heat per lb. than the low pressure steam originally used for heating ; against this we can assume that the high pressure boilers will be more efficient than the low pressure plant they are replacing.

TABLE LXXV. HEAT PUMP WHEN $T_1 = 100°$ AND $T_2 = 34°$ F.

EFFICIENCY RATIO PER CENT.	EFFICIENCY PER CENT.			BTU PUMPED PER H.P.	TOTAL BTU PER 1,000 LB. STEAM	POUNDS OF STEAM REQUIRED	PER CENT. COAL BURNT AT EFFICIENCIES OF	
	AS HEAT ENGINE	AS REFRIGERATOR	AS HEAT PUMP				70 PER CENT.	78 PER CENT.
100	11·81	638	738	—	—	—	—	—
No heat pump			—	—	970,000	100,000	100	
40	4·73	255	355	9,040	1,934,000	50,100		58
50	5·81	319	419	10,670	2,097,000	46,250		54
60	7·09	382	482	12,270	2,257,000	42,950		49
70	8·24	446	546	13,900	2,420,000	41,300		48
80	9·45	510	610	15,530	2,583,000	37,550		43

771. HEAT PUMP ELECTRICALLY DRIVEN. The position when a heat pump is used and is driven by bought electricity must be considered. There are some circumstances where such an installation would be valuable. If the electricity is produced from water power then the heat pump greatly increases the heat available from a given amount of electricity. If the electricity is produced by burning coal then it might seem truly Gilbertian to burn coal in a power station, put half the heat of the coal into the river, convert one quarter of the coal heat into electricity, send the electricity along a cable to a factory or building to drive a heat pump which will recover from the river the heat that the power station poured into it. Nevertheless that is the scheme that is being seriously considered for a large building in central London.

We need not look into the hydro-electric proposition ; that is purely a matter of cost. Such an electrically-driven installation is working in Zürich, taking its heat from the river Limmat which flows out of the Lake of Zürich.

What would be the fuel consumption of a heat pump driven by electricity taken from the British grid ? Let us take an ordinary steam heating system. If its boiler efficiency is 70 per cent. and its distribution efficiency is 90 per cent., its overall efficiency will be 63 per cent. This means that in order to put 1 Btu into a room 1·6 Btu must be liberated in the boiler furnace. The average coal consumption of the grid in 1957 was 1·4 lb. coal/kWh delivered to the customer. Therefore to put 1 Btu of electrical energy into a room by means of an electric radiator will call for the burning of $\frac{10{,}500 \times 1\cdot4}{3{,}415} \times 4\cdot3$ Btu of coal-equivalent in the power station. So that to equal, in coal consumption

alone, a direct coal fired heating system, the efficiency of an electrically-driven heat pump taking power from the grid must be $\frac{4 \cdot 3}{1 \cdot 4}$ 100 = 307 per cent. In order to pay for all the first cost, maintenance and depreciation, it is probable that the heat pump efficiency will have to be nearer 700 per cent.

How then can an eminent man seriously propose to install a plant which cannot possibly have such an efficiency ? The reason is simple. Almost the whole plant would have to be there anyhow. The building is to be cooled in summer and will have a massive refrigerating plant. This could be used in winter as a heat pump to extract from the Thames the heat that Battersea, Fulham and Lots Road so prodigally pour into it.

772. HEAT PUMP EFFICIENCY. The efficiency of the heat pump cycle depends on the difference between the hot temperature and the cold temperature, and we have seen that using ammonia between 100° F. and 34° F. the cycle efficiency is just under 750 per cent. The overall thermal efficiency depends on the efficiency ratio of the machine. What is the efficiency ratio of a heat pump likely to be ? A 50 kW refrigerator in the author's factory had an efficiency ratio of 41 per cent. Haldane in his paper read in 1930 quoted refrigerator efficiency ratios of between 55 per cent. and 65 per cent. There would seem to be no reason why the efficiency ratio of a heat pump should be any greater than that of the same machine running as a straight refrigerator. We can therefore assume that the efficiency ratio will lie between 40 per cent. and 60 per cent., depending on the size of the plant.

There are certain practical considerations that must be borne in mind. In a refrigerator we pour cold water through the compressor cylinder jackets and throw the heat away. We do not lag the hot-end pipes but we encase the cold-end pipes in 3 in. or 4 in. of cork. In a heat pump we must do just the opposite. We must carefully hoard the heat from the compressor cylinder walls. Inside the engine room we must carefully lag both the hot and the cold pipes to prevent shortcircuiting of the heat we are trying to pump. We should draw the air into which we are trying to pump heat through the engine room so that it can pick up the losses that will occur in the engine and compressor. In this way it may be possible for a heat pump to improve a little on the efficiency ratio of the refrigerator.

773. ECONOMICS OF THE HEAT PUMP. Now it is all very well to be technically correct, but we must be economically correct before we embark on a large heat pump installation. If we work out the coal saving that is secured by the use of a heat pump under the conditions shown in Table LXXV with an efficiency ratio of 60 per cent., we find that were the heat pump to be working 70 hours a week for 25 weeks the coal saving would be 4,400 tons a year, and if the heat pump worked for 150 hours a week for 30 weeks a year the saving would be 12,500 tons a year. If we take coal at 100*s*. a ton the money savings become £22,000 and £62,000 respectively. This must be capitalised, at whatever percentage is appropriate to the financing policy of the concern, and compared with the probable cost of the plant which must include the cost of the heat pump installation and the extra costs that high pressure boilers call for over a low

pressure installation. It would appear unlikely that the heat pump would pay unless the plant can work for three shifts, and it would probably never pay to pull out a good plant and substitute a heat pump. But if a new plant is being considered, or if there is already a refrigerator at work the circumstances are much more favourable.

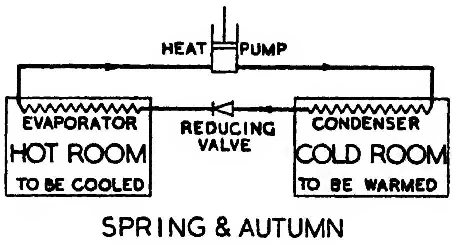


FIG. 423. HEAT PUMP IN SPRING AND AUTUMN

774. FAVOURABLE CONDITIONS. We have so far considered in a little detail a straight heat pump pumping heat from the local river or from the air. There may be much more favourable conditions in some places. There are some factories where rooms or processes have to be cooled and where others have to be heated. Such a condition is shown in Fig. 423 where a heat pump is connected between two rooms, and pumps heat from a hot room into a cold room. This shows the system as it would operate in spring and autumn.

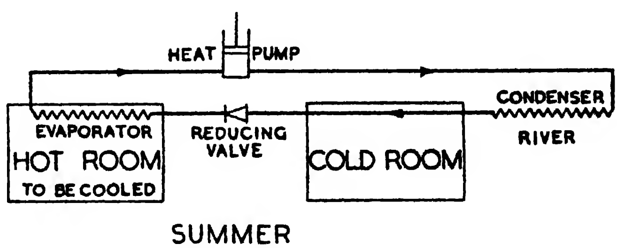


FIG. 424. HEAT PUMP IN SUMMER

In summer, Fig. 424, the cold room will be quite warm enough and will not require heating, but the hot room will be even hotter than usual and will require much cooling. So in summer the heat pump will operate as a straight refrigerator, dissipating the heat that it pumps from the hot room into the river.

In winter, Fig. 425, the hot room will be just comfortable and the cold room will be very cold. In this case the heat pump will pump heat out of the river into the cold room. In all three cases it is assumed that the factory absorbs in its process the whole of the exhaust from the engine that drives the heat pump.

A particularly suitable condition for an installation of this kind would be where a go-ahead municipality ran a skating rink alongside the public wash-house.

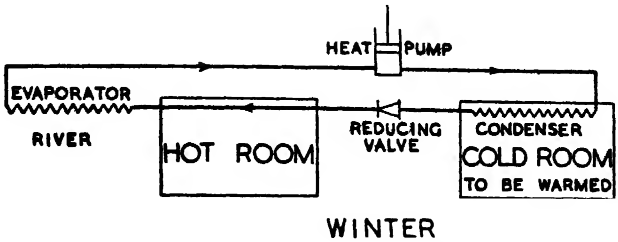


FIG. 425. HEAT PUMP IN WINTER

775. WARM EFFLUENTS. There is still another circumstance which obtains in those factories where there is a large quantity of warm effluent, effluent that is too cool for use direct as a heating medium but which is well above the average temperature of the local river. Let us assume a factory where there is a large quantity of effluent at 75° F. This means that we can raise T_1 from 34° F. to 60° F. The cycle on which a heat pump could operate with such an effluent is shown in Fig. 426. It is clear that a very large gain has been secured. Not only is there much less difference between T_1 and T_2 but the wire-drawing triangle A B E and the superheat triangle C C′ D have been nearly quartered. The cycle efficiency of the heat pump is now 1281 per cent. and Table LXXVI shows what could be expected at various efficiency ratios. The economics are now very much more favourable and the heat pump is definitely in the running under such conditions. Indeed if a

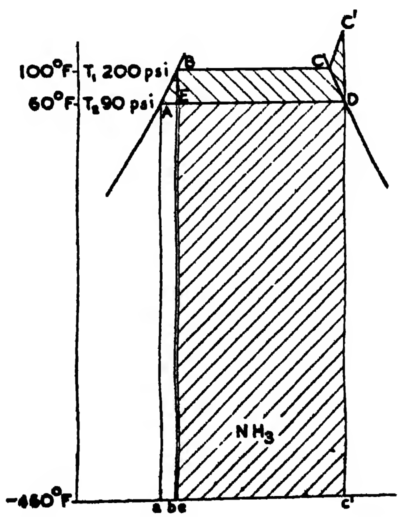


FIG. 426. HEAT PUMP CYCLE WITH FAVOURABLE CONDITIONS

factory has plenty of heating to do and plenty of useless warm effluent it may well be that it would pay both financially and from the national point of view to buy current from the grid to run a heat pump.

Urban sewage is generally warm. It is indeed possible that it is warmer on average in winter than in summer. So here is another possible application for a heat pump in a go-ahead municipality or borough.

TABLE LXXVI. HEAT PUMP WHEN $T_1 = 100°$ AND $T_2 = 60°$ F.

EFFICIENCY RATIO PER CENT.	EFFICIENCY PER CENT.			BTU PUMPED PER H.P.	TOTAL BTU PER 1,000 LB. STEAM	POUNDS OF STEAM REQUIRED	PER CENT. COAL BURNT AT EFFICIENCIES OF	
	AS HEAT ENGINE	AS REFRIG-ERATOR	AS HEAT PUMP				70 PER CENT.	78 PER CENT.
100	7·08	1,181	1,281	—	—	—	—	—
No heat pump			—	—	970,000	100,000	100	
40	2·83	474	574	14,610	2,491,000	38,950		45
50	3·54	591	691	17,390	2,789,000	35,850		41
60	4·25	709	809	20,600	3,090,000	31,400		37
70	4·96	827	927	23,600	3,390,000	29,500		34
80	5·66	955	1,055	26,850	3,715,000	26,950		31

776. FACTORY REFRIGERATION. There are many factories, food and chemical, that require refrigeration, sometimes on a really big scale. No factory should ever embark on refrigeration without investigating whether it can do some heat pumping. We have seen in Section 629 how the heat taken out of the bottled beer in a brewery can be used to heat the mashing water. In the same way the heat taken out of an ordinary meat or fish cold store can be used to heat the offices with very little cost and ingenuity. It must not be forgotten that the heat pump efficiency is 100 + the efficiency of the refrigerator, because in addition to the refrigeration heat we use the power heat as well.

777. THE B.P.E. CYCLE. During the last eighty years cycles making use of Boiling Point Elevation have occasionally been discussed, and some works' locomotives were actually built and run by Honigman at Aachen, about 1885.

The heat pump works over a temperature range of roughly 35° F. to 120° F.

The thermo-compressor works over a range of approximately 200° F. to 240° F.

The B.P.E. cycle can work over a range of possibly 150° F. to 400° F.

It is possible that the B.P.E. cycle is quite unworkable, and the excuse for discussing it here is that there may well be undiscovered or undeveloped thermal improvements waiting round the corner if we could only find them, and the more people there are looking for them the sooner will they be found.

778. BOILING POINT ELEVATION. Before reading this the reader is asked to re-read Sections 3 to 15. When a solid is dissolved in water there are fewer water molecules per unit area of liquid surface than there are in pure water. In order that the solution may exert a sufficient vapour pressure to boil, the water molecules in the solution must have a greater energy than in pure water to compensate for their reduced number. The greater the number of dissolved molecules the greater must be the extra energy in the solution's water molecules to provide a sufficient vapour pressure to permit boiling. It follows therefore that a solution of a solid in water will boil at a higher temperature than the water boiling temperature appropriate to the pressure on the solution. There is a Boiling Point Elevation, or B.P.E., for every particular solution at any particular concentration. The B.P.E. varies with the pressure as well as with the concentration; the higher the pressure the greater is the B.P.E.

The B.P.E. should be predictable from the molecular concentration of the dissolved solid. This is only true of extremely dilute solutions of solids that do not ionize. More concentrated solutions, even of solids like sugar that do not ionize, depart greatly from prediction.

For example, a solution of 16 per cent. by weight of KOH contains 3·4 gram molecules per litre. A 16 per cent. sugar solution contains ·56 gram molecules per litre. The molecular concentration of KOH is $\frac{3{\cdot}4}{{\cdot}56} = 6$ times that of a sugar solution of similar strength. The B.P.E. of the caustic might be expected to be 6 times that of the sugar solution. Actually it is about 20 times.

Here are some examples of B.P.E. at high concentrations :—

Pressure	*Atmos.*			
Concentration	70 per cent.		80 per cent.	
	B.P. °F.	*B.P.E.* °F.	*B.P.* °F.	*B.P.E.* °F.
$C_{12}H_{22}O_{11}$	221	9	231	19
NaOH	358	146	405	193
KOH	441	229	549	337

The steam that comes off an 80 per cent. caustic potash solution boiling at nearly 550° F. is only at atmospheric pressure and can only therefore condense and give up its heat at 212° F. This is the reason why solutions having very high B.P.E. cannot be evaporated satisfactorily in multiple effect evaporators, or why the number of effects may be limited—see Section 492.

779. CONDENSATION. Condensation is usually looked upon as a process where the latent heat in a hot vapour flows into a cooler body. This is the mechanism of condensation in a surface condenser. But in a jet or contact condenser the mechanism is simply one of equalisation of vapour pressure as was explained in Section 15. Now in an 80 per cent. caustic potash solution under atmospheric pressure the vapour pressure of 14·7 psi.a. is not reached until the temperature has been raised to nearly 550° F. It follows therefore

that steam cannot exist in such a solution until the solution has reached its elevated boiling temperature. From this it again follows that steam introduced into such a concentrated solution that is below its boiling temperature must condense although the solution is at a much higher temperature than the steam. The vapour pressure of the steam is higher than the vapour pressure of the solution. Consequently there will be a flow of energy from the steam into the solution and the steam will condense in the very much hotter solution.

The truth of this can be confirmed by the simplest experiment. If water be boiled and the steam bubbled into a strong caustic solution in a glass boiling tube the steam bubbles can be seen condensing in the caustic solution. A thermometer in the steam space of the kettle or flask and another in the caustic solution will clearly show that steam can readily condense in a solution much hotter than itself. Heat thus seems to be "passing of itself from a colder to a hotter body".

The vapour pressure of the steam is higher at 212° F. at atmospheric pressure than that of an 80 per cent. caustic potash solution at 400° F. The pressures will reach equilibrium by the passage of energy from the steam into the solution which will be accompanied by condensation of the steam, the transfer of latent heat to the solution, a rise of temperature of the solution and a dilution of the solution.

Now this appears to contradict the second law of thermodynamics as stated by Clausius. This is not really so. Heat is not "passing of itself". Heat can only pass in this way when the condensing steam is diluting the caustic solution. The process will eventually fizzle out and come to rest.

In the author's factory there is a piece of plant at work called the "vapour melter". The process is the dissolving of washed raw sugar in water. Sensible heat and heat of solution have to be added. The cool sugar-water slurry is pumped through a jet condenser attached to the vapour outlet of a vacuum pan and the vapour condenses in the slurry and supplies the heat required. The B.P.E. of the solution, assuming all crystals dissolved, is about 4° C. The partly dissolved slurry frequently leaves the plant 1° C. or 2° C. hotter than the vapour.

780. CAUSTIC CONDENSER. We can now see that we have a method of stepping up heat in a dramatic manner. If we have vapour at 24 in. vacuum at 140° F. we can condense this in a strong caustic solution that can have a temperature of 300° to 400° F. There is of course a serious snag. Condensation brings dilution with it. If we went on with our condensation process the caustic solution would soon have such a dilution that its B.P.E. would drop and the whole process would break down. It is therefore necessary to pass the caustic solution continuously through an evaporator to maintain its concentration. As the solution has a very high B.P.E. it will be necessary to use very high potential heat to do this concentration. It might therefore be thought that the upgrading that had been done in the condenser would be counteracted in the concentrator. It will be seen that by means of suitable heat exchangers this difficulty can be circumvented.

In Honigman's Aachen locomotives only half the cycle took place in the locomotive. They operated as follows: The "boiler" was a water jacket surrounding a vessel containing hot concentrated caustic. The water in the

jacket was under considerable pressure and was hot. When the regulator was opened, steam, flashed off the water, passed through the locomotive cylinders and the exhaust steam was bubbled into the caustic. This increased the temperature of the caustic which boiled the water. As the temperature of the caustic rose the pressure in the boiler increased until it reached blowing-off point. Some of the pressure was then blown off (apparently they had not thought of simply exhausting to atmosphere) and the locomotive run again. The locomotive was then taken to an evaporator plant where the caustic was reconcentrated.

781. SIMPLE B.P.E. CYCLE. Fig. 427 shows a simple industrial operation in which a product is evaporated under a vacuum of 24 in. by means of steam at 5 psi.g. which has been reduced from the boiler pressure of 45 psi. The vapour from the evaporator is condensed in a jet condenser and the condenser water will have a temperature of about 135° F. If there is a use for large quantities of heat at this low potential, well and good. If there is no use for it (and it can only be used for preliminary heating of liquids or for space heating) the heat in the condenser water must be rejected.

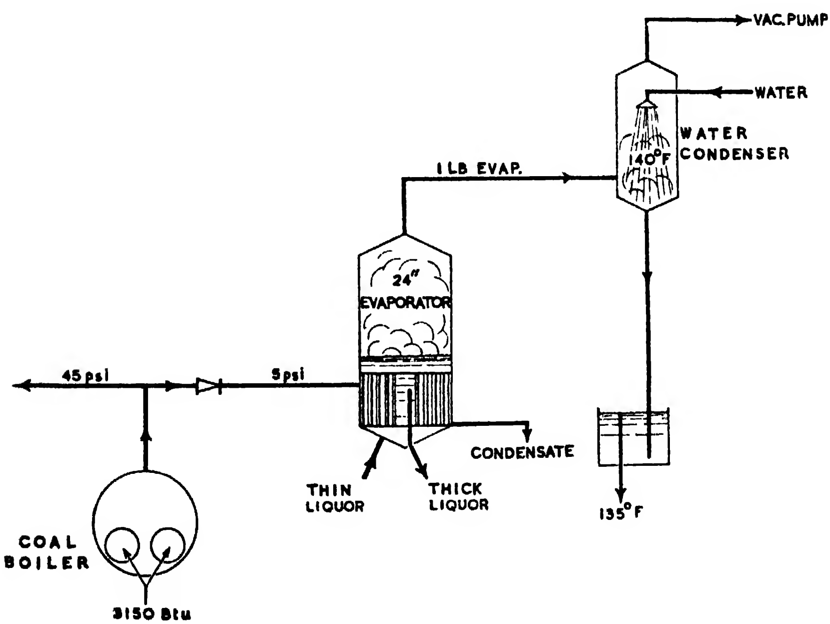


FIG. 427. SIMPLE EVAPORATING PROCESS

Fig. 428 shows how the same process might be carried out using caustic as the condensing medium. The vapour from the evaporator goes into the condenser where it condenses in the caustic solution, adds its latent heat to the caustic and dilutes the caustic. The caustic is drawn from the atmospheric

tank of the caustic condenser and is pumped into a caustic concentrator working under a pressure of 5 psi. If the caustic solution has a concentration of about 50 per cent. its boiling point under 5 psi will be between 300° and 310° F. The hot caustic then passes through a heat exchanger where it can heat water under pressure to at least 250° F. or can raise steam at about 10 psi. Heat at this potential will be useful in any factory. The cooled caustic then passes into the condenser and the cycle repeats. The vapour off the caustic concentrator is at 5 psi and is used to heat the process evaporator instead of live steam.

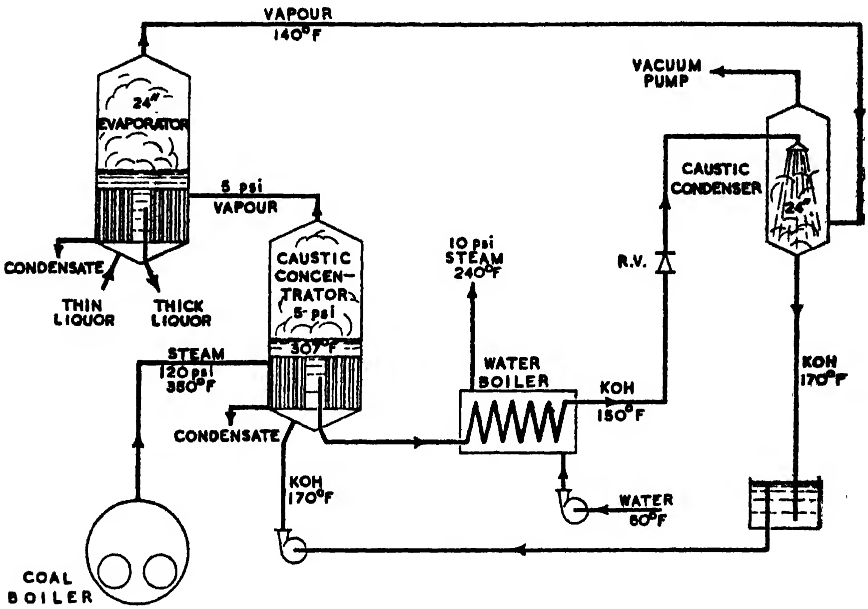


FIG. 428. SIMPLE B.P.E. HEAT RECOVERY AND UP-GRADING CYCLE

We see that in Fig. 427 the whole of the heat leaves the evaporation process at 135° F., whereas in Fig. 428 the whole of the heat is rejected from the evaporation process at 240° F. The caustic cycle has stepped up the heat by over 100° F.

782. ADVANCED B.P.E. CYCLE. Fig. 429 shows an arrangement, suggested by the author's brother in 1932, whereby the heat rejected from the process can appear as steam at 45 psi or at any higher pressure that may be found workable (if the cycle is workable at all). The vapour from the evaporator is condensed in a caustic condenser. The caustic is drawn from the condenser atmospheric tank by a feed pump and pumped at about 90 psi through a heat exchanger where it picks up about 170° F. It then goes into the coal-fired boiler where it receives the addition of about 1,320 Btu per lb. of evaporation in the evaporator. It then passes through a surplus valve into the

flash vessel at 45 psi. The water that it picked up in the condenser is flashed off and goes to the factory as 45 psi steam. During flashing the caustic loses about 23° F. It then goes through the heat exchanger where it parts with some 160° F. to the caustic that is on its way to the boiler. The part-cooled caustic then goes into a water boiler where it loses 33° F. and generates the steam necessary for the process evaporation. It then enters the caustic condenser and the cycle repeats.

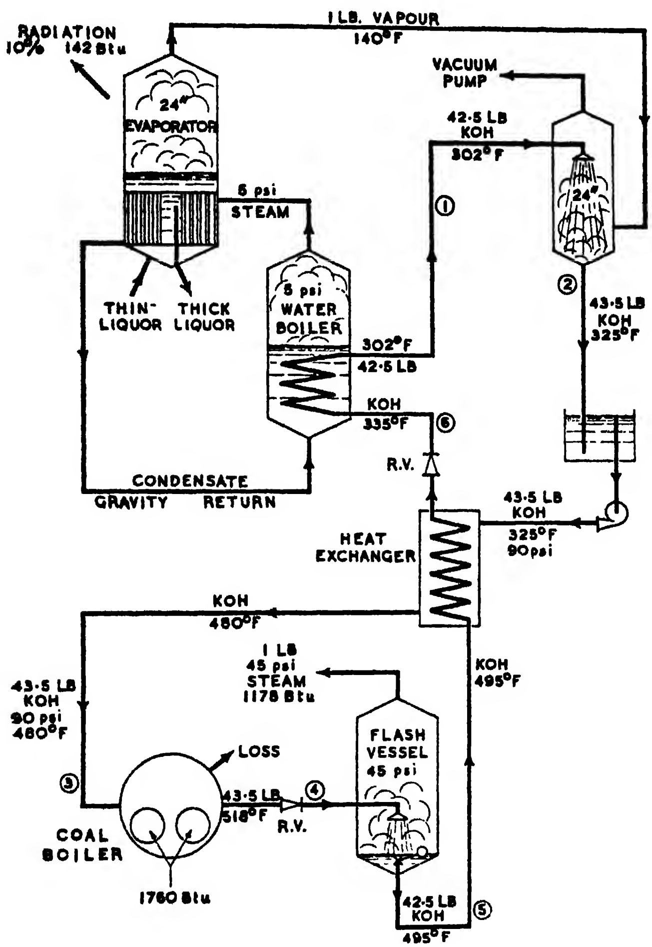


FIG. 429. ADVANCED B.P.E. CYCLE

The whole thing bristles with difficulties. The quantities on Fig. 429 are guesses and were taken merely to show the practicability of the thermal cycle. For every 1 lb. of evaporation some 43 lb. of caustic solution must be circulated

and must be pumped up to about 90 psi. This will require about 9 H.P. per 1,000 lb. of evaporation per hour. This compares well enough with the heat pump but is not so very simple because the material to be pumped is hot, very concentrated caustic. The output steam at 45 psi will almost certainly be alkaline and may be useless for process purposes. The whole plant must be made of corrosion-resisting metal. The gain, on the other hand, is potentially great. Comparing Fig. 427 with Fig. 429 we see that to provide the same 45 psi

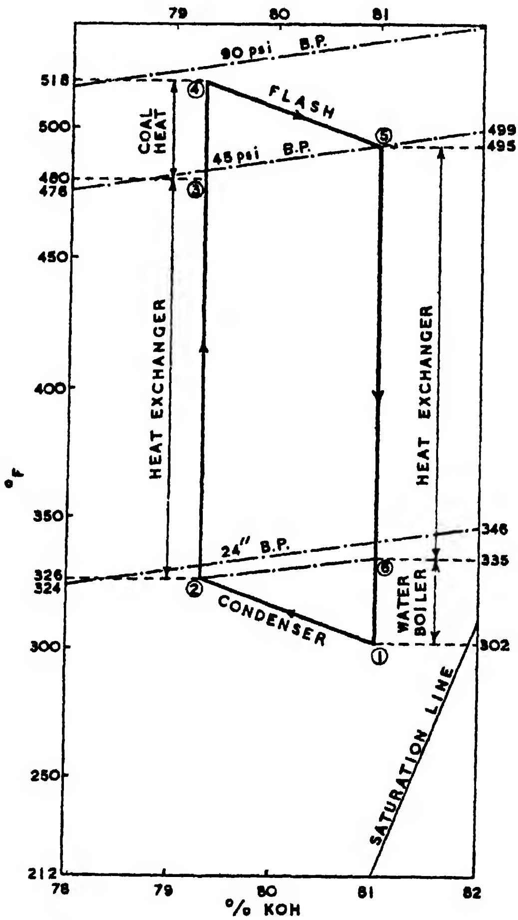


FIG. 430. SOLUTION STATE DURING CYCLE SHOWN IN FIG. 429

steam to process, to do the same evaporation, allowing a 5 per cent. radiation loss from the plant in Fig. 427 and a 10 per cent. radiation loss in Fig. 429 requires over 3,000 Btu in Fig. 427 and only 1,760 Btu in Fig. 429. The caustic cycle is almost twice as good as the straight cycle.

Fig. 430 is a diagram which shows the goings on within the caustic solution. This gives the state of the solution at each point in the cycle. The figures in circles correspond to the similar figures in Fig. 429. This diagram enables us to keep control of things. For example the bottom right hand of the diagram shows the caustic at its coolest and most concentrated. This point must be kept well clear of the saturation line lest crystallisation should start. The top left-hand point in the diagram shows that in order to reach the temperature that is wanted in the coal boiler the pressure must be at least 90 psi lest the solution should boil in the boiler, which is not wanted, although the flash vessel could easily be replaced by a steaming coal-fired caustic boiler. As the reduction of entrainment is of the utmost importance it is probable that the flash vessel will be better.

This particular arrangement may be quite impracticable, but the possibilities are so great that it is likely that some such cycle, on a workable plan, will soon be discovered.

CHAPTER 26

THE ICE-WATER-STEAM SUBSTANCE

Here's to old Adam's crystal ale,
Clear sparkling and divine,
Fair H_2O, long may you flow,
We drink your health (in wine).
O. HERFORD. *Toast.*

The scientific composition of water is of no practical importance for the correct application of steam technique. This chapter is only added for the interest of those who may wish to know some of the intricacies of chemical and physical mechanism that lie hidden in snow, in our rivers and wells, or even in our fogs. The author is neither chemist nor physicist, he is not even an engineer, and the following information has been culled from all kinds of sources.

783. HYDROGEN HYDROXIDE. Chemically, water for long was looked upon as being a simple oxide of hydrogen—H_2O. Its chemical behaviour however leads us to believe that it is really an hydroxide of hydrogen—H (OH). The hydrogen H of water can combine with a non-metal or combination of non-metals to form an acid. The hydroxyl OH part of the water can combine with one of the light metals to form an alkali. Acids and alkalis are the most active and important chemical substances and they all spring from water's loins. Water is the basis of the whole of our terrestrial chemistry and carries the two most important active particles or ions—the hydrogen ions, or acid-producers, and the hydroxyl ions, or alkali producers.

784. THE HYDROLS. Physically, water was until recently looked upon as being composed of single simple H_2O or H (OH) molecules. More accurate determinations of the physical properties of water indicate that water must consist of a mixture of at least two, probably three, possibly more molecular arrangements. It is suggested that water really consists of a solvent, di-hydrol $(H_2O)_2$, carrying in solution steam molecules, hydrol (H_2O), and ice molecules, tri-hydrol $(H_2O)_3$.

At low temperatures the steam or hydrol molecules all disappear, and at higher temperatures near the boiling point the ice or tri-hydrol molecules disappear. This theory is said to account satisfactorily for several of the observed phenomena of water, namely, the changes of specific heat, the changes of colour, the scattering effect on light, the change of density, etc.

There is some evidence of the existence of very long chain molecules forming a network at the " skin " or surface of water. This is said to account for some surface tension effects.

785. STEAM. The single unattached molecule, hydrol, makes up the steam substance. The hydrol molecule is pictured as being of tetrahedral shape, that is a three-sided pyramid standing on a triangular base. The pictured structure of a molecule is imaginary but certain conventional mind pictures are useful.

For example, an atom is considered to consist of a positively charged nucleus with negatively charged electrons revolving in orbits, like planets, around it. This picture of the atom means that it is a kind of very empty fuzzy sphere.

When two atoms of hydrogen combine with one atom of oxygen to form hydrol, it is thought that the oxygen nucleus remains at the centre and that the hydrogen nuclei place themselves near the " surface " at two of the points of an imaginary tetrahedron, as shown in Fig. 431.

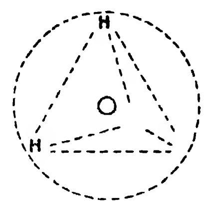


FIG. 431. NOTIONAL DIAGRAM OF STEAM MOLECULE

786. **ICE.** Whenever a group of molecules is so cooled that the relative motion between the molecules is overcome by their mutual attraction, they fasten themselves together, the positive hydrogen point or points of one molecule linking with the vacant negative point or points of another.

When the molecules have linked themselves together in this way they form a definite structure or crystal lattice which is called Ice. There are obviously a number of different ways in which water molecules can link themselves together symmetrically.

Eight different kinds of ice have been reported by various workers. These are known as Ice–I, Ice–II, etc. The denser ices only occur under very high pressures and need not concern us, except to note that they exist.

If we assume that ice–I is a structure where each H_2O molecule is in contact with four others, it is clear that the structure must be more open than were the molecules a pile of slippery spheres where, if stationary and regular, each sphere would be in contact with 12 neighbours.

When the molecules are arranged in an ice structure they are locked tightly together and can only vibrate, they cannot move freely or independently. Under a pressure of 13 or 14 tons/sq. in. the structure of ice–I collapses and one of the other molecular arrangements appears—the heavy ices, ice–II or ice–III.

787. **MELTING.** When heat is added to ice the molecules vibrate more strongly until some of them break their fetters. These molecules are probably still attached to some neighbours, but their structure has ceased to be rigid and they can slither about inside the openings and crevices of the ice structure. When sufficient heat has been added the structure breaks down completely, but this does not mean that the molecules have broken all their bonds.

As the temperature of water increases, a greater intermolecular movement will occur and, as a result of this extra activity, the molecules should be a little further apart. Consequently there should be a steady expansion of water from the freezing or melting point upwards. This however is not the case. The density of water increases from the melting point 32° F. to 40° F. after which a steady decrease in density occurs. This can be readily explained if we assume that when ice has broken down into water there are still large numbers of molecules that are sufficiently bonded to their neighbours to occupy more space than that occupied by water. Although the increase of temperature from 32° to 40° does cause an expansion, there is a contraction due to the breakdown of bonded groups of semi-ice that more than compensates, and causes the volume to contract until 40° F. is passed.

788. TRIHYDROL. Ice–I is generally looked upon as being $(H_2O)_3$, though some authorities favour $(H_2O)_4$ or $(H_2O)_6$, one enthusiast even going so far as to suggest $(H_2O)_{23}$. Whatever the molecular linkage, it is known that ice has a crystalline structure—this can be seen very beautifully in certain kinds of snow, and hoar frost is obviously crystalline, but the crystal structure is not so obvious in an icicle. The structure must be an open one because the density of ice is only about nine-tenths that of water.

789. WATER—DIHYDROL. If water is composed of a solvent, dihydrol $(H_2O)_2$, containing more or less hydrol (H_2O) and trihydrol $(H_2O)_3$ in solution, we must assume that each dihydrol molecule consists of a pair of active hydrol molecules tied together. Each dihydrol molecule group might be like Dr. Doolittle's immortal animal, the "Pushmepullyou", or like a pair of independently-powered, independently-steered motor cars connected by a very powerful magnet. An occasional collision will break the connection when there will temporarily be two hydrol molecules. If the temperature is low enough, that is to say if the molecules are going slow enough, these single hydrol molecules will find mates that they will hungrily clasp. If the temperature is approaching the freezing point, single hydrol molecules will attach themselves to dihydrol molecules to form ice molecules.

It is almost certainly wrong to look upon water as a collection of slippery spheres. It is a complicated, jumbled mass of double molecules with more or less single and triple molecules, with many empty spaces and many local areas of traffic congestion. The whole thing is seething and milling round with molecules being knocked apart here and joining together there. Most of the movement of water is probably vibrational or rotational; only an occasional single hydrol molecule or pair of molecules will actually be swimming through the jumble.

790. BOILING. When water approaches boiling point quite a number of dihydrol molecules have been knocked apart into single hydrol molecules. The single molecule is much better placed to proceed on short sharp excursions through the hurly-burly, and most of the actual translational movement will be vested in the single hydrol molecules. The hotter the water the more violent is the movement, whether this movement be vibrational, rotational or

translational. The mutual attraction of the molecules is still sufficient to keep them in mutual semi-contact, but some near the surface, are so jostled that they jump out as explained in Section 6.

When actual boiling temperature is reached the movement is such that any further activity ejects the single molecules, breaks the bonds of the double dihydrol molecules and the loss of energy due to the escapists keeps the temperature from rising further.

The theoretical boiling temperature, due to the pressure under which boiling is to take place, together with any hydrostatic pressure, may be appreciably lower than the temperature actually needed to produce boiling, because there is an additional force to be overcome, namely surface tension. A minute bubble has a very large surface/volume ratio and the surface tension of its envelope is a material force. As the bubble increases in size the surface/volume ratio rapidly decreases and the surface tension component resisting evaporation diminishes, so that a steam bubble has an inherent explosive tendency.

791. ICE—WATER—STEAM. Cold ice is a rigid pattern of locked molecules in which by far the most important force is the attraction of the positive hydrogen points on the shell of the molecule for the negative areas on the adjacent molecules. These locked molecules can only vibrate.

Water is a random jumbled arrangement where the principal force is still the mutual attraction of the positive and negative charges and where most of the molecules are linked in pairs, but where at low temperatures there are plenty of triplets and at high temperatures there will be an increasing number of single steam molecules in solution. There is much vibratory movement, a lot of rotary movement and a little translatory movement. At the water surface there may be a network of very long chain molecules forming a " skin ".

Steam is a disorganised system of single molecules where the mutual attraction between the molecules has been completely broken down and where the great bulk of the energy is in the form of translational motion, although there is probably still much rotational movement as well.

792. LATENT HEAT OF FUSION. When ice melts its temperature remains constant while it absorbs and stores the heat energy by breaking the bonds that lock the structure together. The loosened molecules retain the bond-breaking energy as vibrational and rotational energy. There can be no rise of temperature because there is no increase in the kinetic energy. The process is comparable to that of the vaporisation of steam described in Sections 6 to 14. The latent heat of the melting of ice is 143 Btu/lb.

793. SPECIFIC HEAT OF ICE. The apparent specific heat of ice varies with the purity of the water that is freezing. As ice forms the impurities become more and more concentrated in minute inclusions of liquid in the ice. The purer the water the smaller are such inclusions and, except for highly academic purposes, need not be considered.

The specific heat of ice varies with the absolute temperature, apparently becoming zero at absolute zero. The specific heat is of no practical importance except near to the freezing point for refrigeration calculations. For practical purposes the specific heat of ice can be taken as being ·5. Table LXXVII gives round figure approximations of a number of published values.

TABLE LXXVII. SPECIFIC HEAT OF ICE

TEMPERATURE ° F.	APPROXIMATE SPECIFIC HEAT
32	·5
− 60	·4
− 165	·3
− 275	·2
− 375	·1
− 400	·05

794. **TOTAL HEAT OF ICE—WATER—STEAM.** If the specific heat of ice varies in accordance with the values in Table LXXVII it will have a total heat, above absolute zero, of 126 Btu/lb. The latent heat of fusion is 143 Btu/lb. so that water at 32° F. has a true total heat of 269 Btu/lb. The sensible heat of boiling water at atmospheric pressure is 180 Btu/lb. and the latent heat of vaporisation is 971 Btu/lb. so that the real total heat of 1 lb. of dry saturated steam at atmospheric pressure is 1,420 Btu. These heat additions are shown on the temperature entropy diagram for ice, water and steam in Fig. 432.

795. **THE UNIQUE PROPERTIES OF WATER.** Water is the basis of all life. All animal and vegetable structures are based on water. The juices and fluids in all forms of life are water solutions.

Water has however other curious properties. One of the most interesting is that water has its greatest density at 40° F. If the maximum density were at 32° F. it is possible that life on earth would be considerably reduced.

Let us consider the freezing of a pond. As the water at the surface is cooled it sinks until the whole water content of the pond is at 40° F. Further cooling of the pond surface causes a slight expansion of the surface water which therefore floats where it is and discourages convection currents. Eventually the top layer solidifies. The water in the bulk of the pond is probably well above 32° F., most of it will be at 40° F. Further freezing can only take place by conduction through the thickening ice layer which acts as an insulator.

When warmer conditions come along the coldest part of the pond, the ice sheet, is the first to receive warmth and it melts. As the melted ice-water approaches 40° it sinks and any water that is below 40° F. rises and is in turn warmed.

Now suppose that the maximum density of water was at 32° F. During cooling the coldest water would sink until the whole of the water in the pond had been cooled to 32° F. Ice formation would then be more rapid, and would be much more permanent. The ice sheet would be much thicker and in a

severe winter many ponds would be frozen solid. During the thaw the upper surface of the pond would be warmed. After the ice had melted the newly melted water would not sink and be replaced by cooler water. The

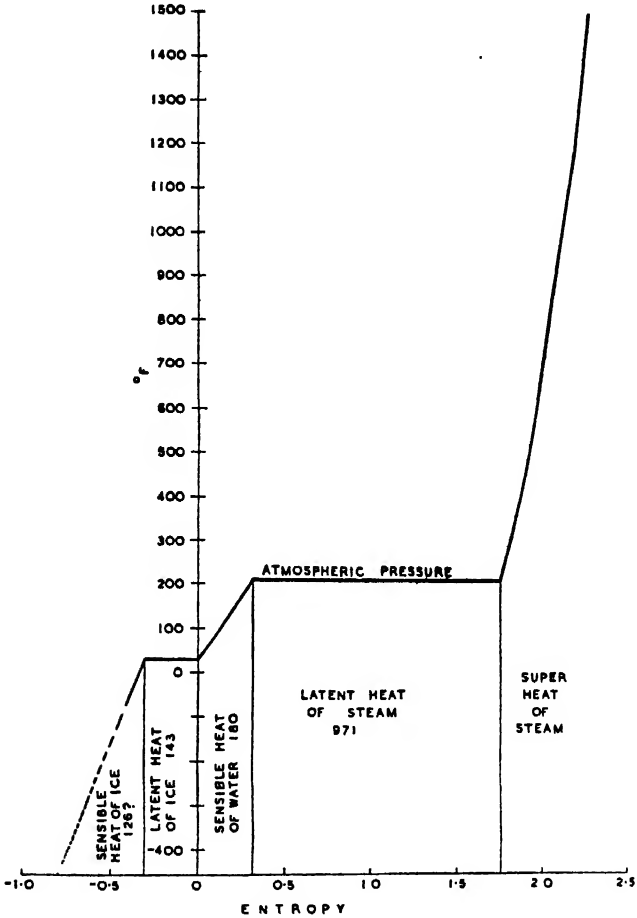


FIG. 432. TEMPERATURE/ENTROPY DIAGRAM OF ICE—WATER—STEAM

newly melted water would have a lower density and would float and some of the deeper ice might never be melted. Consequently when another frost came along the whole pond would be cooler and would freeze more readily. Many ponds that at present get a mere skin of ice on their surfaces would be frozen solid and the life in them would be destroyed.

From a thermal point of view water has superb qualities. It is the supreme heat carrier. It has a very high specific heat and a very high latent heat. It is therefore able to hold a great amount of heat in a very small bulk or weight.

Water is stable and non-corrosive. It is abundant, tasteless and odourless. In Table LXXVIII are some of the physical constants of some other liquids for comparison with water. Of all the liquids that are reasonably stable and cheap only alcohol and glycol would seem to be suitable for use in heat engines, alcohol for low temperature ranges and glycol for higher temperatures. Carbon tetrachloride clearly has claims for use as a thermostatic liquid.

TABLE LXXVIII. PHYSICAL PROPERTIES OF LIQUIDS

	ATMOSPHERIC BOILING POINT ° F.	SPECIFIC HEAT BTU/LB.	LATENT HEAT BTU/LB.	TOTAL HEAT OF VAPOUR AT ATMOSPHERIC BOILING POINT BTU/LB.
Water	212	1·0	971	1,151
Acetone	133	·51	223	275
Aniline	356	·51	187	352
Alcohol—Ethyl ..	164	·68	368	458
Alcohol—Methyl ..	148	·61	473	544
Benzene	176	·47	169	237
Carbon Tetrachloride ..	170	·20	84	112
Chloroform	142	·24	100	132
Cresol	394	·55	181	380
Glycol	387	·58	344	551
Mercury	675	·033	122	146
Toluene..	231	·53	156	261

796. WATER AS A HEAT ENGINE FLUID. Figs. 22 and 24 in Sections 83 and 88 show water as a very imperfect fluid for a heat engine. We know from Carnot's theorem (Section 91) that the whole heat addition should be made at the highest temperature of the cycle. In all water-steam cycles that aspire to efficiency this is far from being the case. In Figs. 22 and 24 practically no heat is added at the highest temperature, yet the superheater, valves, turbine, etc., all have to be made to stand this high temperature.

In Fig. 24, Section 88, it was necessary to try to make the left-hand line of the diagram vertical by bleed heating. This is because water has such a high specific heat.

The sharp superheat " horn " has to be added because the latent heat of steam changes so much with temperature.

In Fig. 433 is a simple water-steam power cycle where the highest temperature is 800° F. at 650 psi exhausting at 28·5 in. vacuum. The cycle is shown as the full line and has a cycle efficiency of 35·6 per cent.

Dotted on the same diagram is the boundary curve of an imaginary liquid called "nectar" which has a very low specific heat. The liquid line AB is so nearly vertical that it would hardly be worth while adopting bleed feed heating.

This imaginary liquid has a latent heat of vaporisation that varies only slightly with pressure so that the saturation line CD is so near to being vertical that no superheat is necessary. This enables almost all the heat to be added along BC at the highest temperature of the cycle.

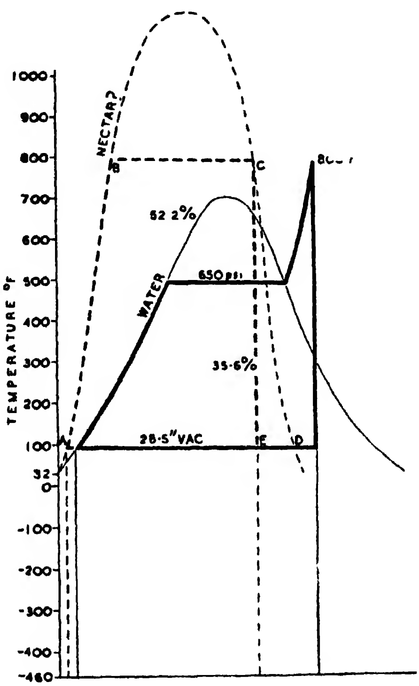


FIG. 433. POWER CYCLES WITH WATER AND IMAGINARY FLUID AT CONSTANT MAXIMUM TEMPERATURES

The work area A B C E is nearly a rectangle and approaches a Carnot cycle. Were such a liquid available the cycle efficiency between 800° F. and 92° F. would be 52·2 per cent. against 35·6 per cent. with water.

Unfortunately up to the present the chemists have not been able to offer the engineers such a liquid.

797. VISCOSITY OF WATER. Water changes in viscosity much more than most people remember and curious flow and heat transfer effects can often be accounted for by changes in viscosity. (See Section 213.) A few viscosity values were given in Section 356. A list of values between 32° and 212° F. is given in Table LXXIX.

798. SOLUBILITY OF GASES IN WATER. The approximate solubilities of Air, Carbon Dioxide, Oxygen and Nitrogen in water are given in Fig. 434.

TABLE LXXIX. VISCOSITY OF WATER

TEMPERATURE ° F.	VISCOSITY	
	CENTIPOISES	LB./SEC. FT.
32	1·79	·001203
40	1·54	·001035
60	1·12	·000753
80	·86	·000578
100	·64	·000430
120	·56	·000376
140	·47	·000316
160	·40	·000269
180	·35	·000235
200	·30	·000200
212	·28	·000190

799. WATER THE SOLVENT. Pure water—a very rare material, is an almost universal solvent. Some very high pressure boilers, especially those of the single tube type, demand a feed water almost free from dissolved solids. Such water can be and is being produced, but it is not the innocent fluid it might seem at first sight to be. It dissolves everything, heat exchangers, feed pumps, etc. The purer the water the more important becomes the pH control. Hot, newly condensed water is particularly prone to dissolve plant. There was probably some CO_2 in the trap in which the condensate was collected and the water dissolved the gas and thus became at birth an excellent solvent for other things, metals in particular.

The distilled water produced by the quadruple effect evaporator described in Section 761, dissolved 15 lb. of steel every week from the plant for over a year before the pH of the distilled water was brought under adequate control. The eventual cure was to inject a little boiler blowdown into the steam spaces of the evaporator calandrias.

Condensate pipe lines are for this reason particularly liable to corrosion from condensate. They are much more likely to suffer than feed water lines where the water has been treated and possibly de-aerated. In the author's factory the copper coils of an old vacuum pan were replaced by a new set of steel coils which were given a very good fall to clear the condensate promptly. Within three years the condensate had cut a wide groove along the bottom of the coils $\frac{3}{16}$-in. deep.

Pure water if it is truly neutral is relatively harmless, but water so often has a chance to take up CO_2, when it at once becomes a potential danger. De-aeration should be done to all pure water at the very earliest possible

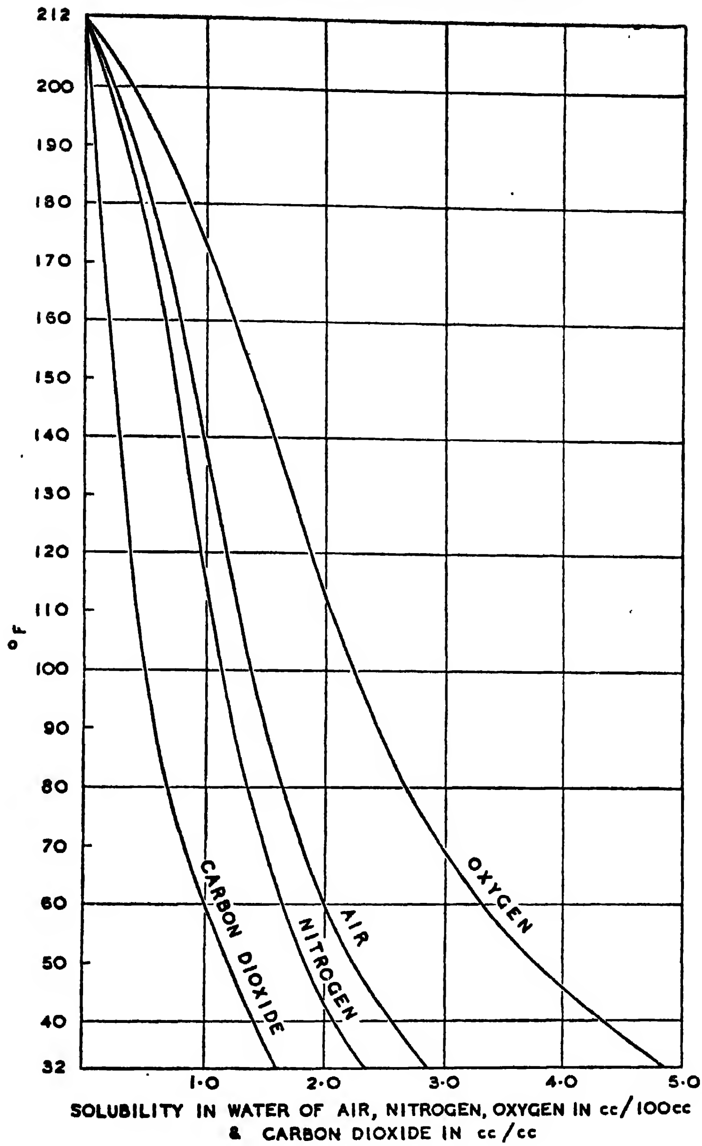


FIG. 434. SOLUBILITY OF GASES IN WATER

moment of its life. Complete de-aeration is a somewhat costly and difficult operation to do by means of heat, or flash, and it may be necessary to adopt chemical means, but the addition of chemicals may add impurities, and the water ceases to be pure.

Pure distilled water will dissolve quite a detectable amount of glass from the bottles in which it is kept. If stored in ordinary glass bottles for three or four

weeks it may dissolve sufficient silica, etc., as to render it unsuitable for ordinary accurate analytical purposes.

800. WATER THE SOLUTION. Natural water is simply a dilute solution of salts and gases. The proportions of the commoner salts found in water vary greatly. It is the composition of the impurities that gives to different waters their particular characteristics.

TABLE LXXX. COMPOSITION OF VARIOUS WATERS

	PARTS PER 100,000				
	BURTON WELL	DUBLIN RIVER	LONDON METROPOLITAN WATER BOARD	LIVERPOOL CORPORATION	IRISH SEA
Total dissolved solids	165·7	31·4	48·8	6·9	3,386·0
Calcium carbonate..	10·90	20·30	19·40	—	4·8
Calcium sulphate ..	111·24	6·36	16·05	3·14	133·2
Magnesium carbonate	30·44	1·29	—	—	Tr.
Magnesium chloride	—	—	4·16	1·43	—
Magnesium sulphate	—	—	·80	·29	—
Potassium sulphate..	1·99	—	—	—	—
Sodium chloride ..	5·57	2·61	1·88	1·14	2,643·9
Sodium nitrate ..	2·81	—	5·36	—	—
Others including iron and silica	1·40	·70	1·00	·39	604·1
	RAIN WATER, LAND'S END	RAIN WATER, LONDON	DEW, ROTHAMSTEAD	SNOW, KEMSING	
Total dissolved solids	42·8	2·76	4·87	4·25	
Organic carbon ..	·131	·383	·264	·306*	
Nitrogen	·054	·258	·297	·003?	
Chlorine	21·8	·50	·53	·55	
Hardness	10·0	1·1	1·9	·95	
	PER CENT. BY VOLUME				
	RAIN WATER	LOCH KATRINE WATER			
Nitrogen	1·308	1·731			
Oxygen	·637	·704			
Carbon dioxide ..	·128	·113			

* There were also 3·0 parts of carbon as visible soot flocs in suspension.

Distilled water is pretty pure. Dew and snow water are not so pure but are still fairly pure. These waters are flat uninteresting drinks. Water with some salts in solution is pleasanter but the most important quality that makes water taste good is the presence in solution of air or CO_2. For this reason newly boiled water is a very poor drink.

It is on the composition of the water used for brewing that the quality of beer largely depends. Most brewers try to treat their water so as to imitate the well water at Burton-on-Trent. Burton water has a most curious composition. It contains an astonishing amount of permanent hardness, and many brewers pass their water over gypsum in an endeavour to raise the hardness of their local water to Burton level. Many brewers, whose natural water contains much temporary hardness, heat all their water to throw down the calcium carbonate. This wastes a great amount of steam. Were they to add a little sulphuric acid, they could convert their carbonate into sulphate for a trifling cost. Although they freely add sulphate in the form of gypsum, they are so nervous of public opinion that they dare not use the rational and economical alternative of converting carbonate into sulphate. This is an excellent example of the way an ignorant public opinion can call for inefficiency.

All sorts of treatment can be given to water to remove undesirables or to add desirables. This book is no place for a discussion on water treatment. There is plenty of literature on the subject and some books on water treatment are listed in Section 805. Table LXXX shows the characteristics of several different waters.

Burton water has clearly percolated through beds from which it has dissolved a lot of things. Liverpool water comes from the mountains of North Wales where it has run quickly down the rocky places into Lake Vyrnwy before it has had much time to do much dissolving. London water however is chiefly Thames and Lea water which flows through flat lands with only a few small hills. Its water has taken a long time to reach it and has had much opportunity to pick up salts.

The rain water at Land's End has picked up a lot of spume from the sea and is anything but pure compared with London rain. On the other hand the carbon content of London rain shows the effect of the smoke in a large city. The impurities of dew and snow are surprising. The knowledge of their true composition would flabbergast the poets.

801. HEAVY WATER. Quite recently it has been found that water contains traces of an oxide of the hydrogen isotope ^2H, or deuterium. Deuterium has an atomic weight of 2, double that of ordinary hydrogen, 1. It is possible by extremely laborious and costly methods to obtain a concentrate of water containing deuterium oxide. This is the so-called heavy water which differs from ordinary water in its physical properties. As far as the practical steam user is concerned, it is, as far as we know, of no importance. It may have great importance in the future in the harnessing of atomic energy.

* * *

EPILOGUE

> Whatever thou takest in hand, remember the end, and thou shalt never do amiss
>
> ECCLESIASTICUS. VII 36. B.C. 180

802. This book, which started as a simple stringing together of a handful of bulletins, had become, long before its end was in sight, a much more ambitious volume. The author does not claim to have covered every use and trick in steam's repertory—there are dozens of processes of which he is entirely ignorant —but he believes that the most important matters have at least been mentioned.

There has doubtless been much ill-balanced emphasis. Some important things have received the lightest of treatment while other trivialities have been heavily laboured. But the heavy hand on the small point has had in view the showing up of the method of attack and the possible road to the solution.

Many of the things described in the book may be of most limited application. Some things, for example the curious machines depicted in Figs. 270 and 303, may never find an economic home in any factory. But the consideration of such things may point a way of doing the job much better in some related way. ALL thermal devices deserve investigation, deserve a careful estimate of their probable cost and deserve a conscientious calculation of the return they may bring. "Remember the end, and thou shalt never do amiss."

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803. BINARY AND OTHER CYCLES. In Section 89 a steam power cycle is shown with a theoretical cycle efficiency of some 46 per cent. This was obtained by raising steam at 1,000 psi, superheating to 900° F., reheating to 800° F. and feed heating by regenerative turbine bleeds. By raising the boiler pressure to over 2,000 psi and superheating to 1000° F. the cycle efficiency can be raised to just over 50 per cent. Now high temperature at high pressure brings in all kinds of engineering and metallurgical difficulties. An examination of Fig. 25 shows us that the high temperature, which brings all these difficulties is only accompanied by a very small heat addition. Most of the heat is added at between 500 and 600° F. If the pressure drop were greatly limited it would be possible to keep the cycle within the boundary curve and therefore approximate to a Carnot rectangle, but much good energy would be lost.

Binary Cycles

Mercury boils at 900° F. under the very modest pressure of 80 psi. At 28·5 in. vacuum mercury vapour has a temperature of 437° F. If therefore, mercury is boiled in a boiler, and the saturated mercury vapour passed through a mercury turbine the mercury vapour can be exhausted into a condenser which can be a water boiler, and can raise steam at 250 psi. This steam can be given some superheat up to say 600° F. from the flue gases of the mercury boiler. Owing to the low latent heat of mercury it is necessary to use about 10 pounds of mercury for every 1 pound of water.

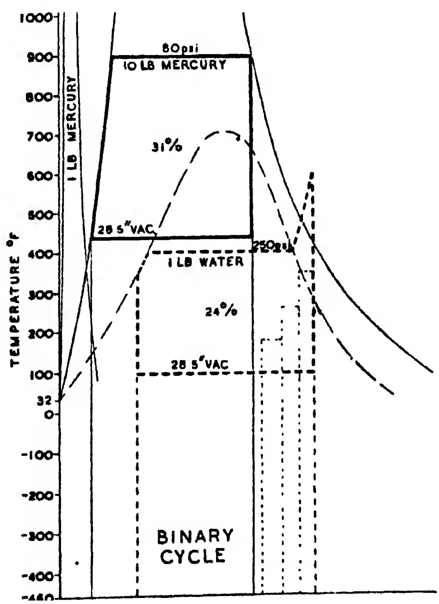

FIG. 435. BINARY POWER CYCLE—MERCURY AND WATER

Fig. 435 shows the double cycle. The slim curve at the left of the diagram is the boundary curve for 1 lb. of mercury. The cycle is shown with 10 lb. of mercury and 1 lb. of water.

The heat balance is roughly this :—

Coal heat into mercury	1,410 Btu.
Coal superheat into steam	115 ,,
	1,525 Btu.

	Btu	Per Cent.
Power available from mercury turbine ..	470	31
Power available from steam turbine ..	370	24
	840	55

Now this 55 per cent. has been obtained without any cycle refinements. 10 lb. of mercury to 1 lb. steam is only approximately the proportion needed. By proportioning the fluids correctly and adding all kinds of heat exchange refinements, the efficiency of the cycle can be raised still more.

This binary cycle clearly has attractions on the face of it. Almost all the heat input is done at the top temperature. At this high temperature the pressure is quite low—only 80 psi.

The disadvantages are somewhat formidable. Mercury vapour is extremely poisonous. Mercury does not wet metal surfaces. The plant is complicated and costly. But several large mercury-steam stations have been working for some years in U.S.A. and have shown very high sustained overall thermal efficiencies. Other fluids might give better results. Diphenyl oxide has been suggested as being more suitable than mercury.

Compression-Condensation Cycles

In Section 88 the cycle using compression to produce an extra gain of cycle efficiency was mentioned. The cycle consists of condensing the exhaust incompletely and compressing the steam-condensate mixture up to boiler pressure. If the steam is saturated initially this gives us the Carnot Cycle. Such a cycle must therefore be worth examining.

As the bald idea of recompressing exhaust steam to boiler pressure appears crazy, it merits a little discussion. If the whole of the exhaust steam from a perfect engine using saturated steam were recompressed to boiler pressure by a perfect compressor driven by the engine, the set would give no power and would work on an ordinary perpetual motion cycle. We can therefore say that exhaust steam recompression for power purposes is no good.

But the compression of low pressure steam so as to raise its temperature to enable it to transfer its latent heat, is another matter. Thermo-compressors for this purpose are in use in industry—see Section 493. The heat pump, Sections 764–776 relies entirely on this principle.

In the compression-condensation cycle this is what the compressor does. The compressor takes a large weight of condensate, containing only sensible heat, and a small weight of steam, containing sensible and latent heat. The condensate is pumped up to boiler pressure at constant low temperature. The steam, on the other hand, increases in temperature as its pressure rises. It consequently transfers its latent heat to the condensate and appropriately condenses as its pressure rises. If the proportions of steam and water at the inlet are well chosen, the high pressure output of the compressor will be water only at the boiling temperature corresponding to the pressure.

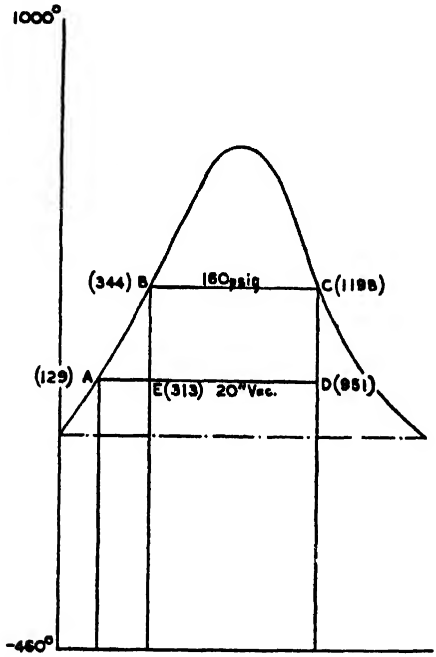

FIG. 436.

In order that the compression-condensation cycle should have the best conditions, the amount of steam in the steam-condensate mixture entering the compressor must be equal to the amount of steam that would be flashed off during the expansion of water from boiler temperature and pressure to the condenser pressure. Any less quantity of steam is simply loss to the cycle —it approximates more to straight condensation. Any greater quantity of steam reduces the usefulness capacity of each pound of steam, but does not theoretically damage the cycle efficiency.

Fig. 436 shows the cycle, compared with an ordinary straight condensing cycle, applied to an old-fashioned mill engine.

Steam at C, 160 psi saturated, is expanded to D, 20-in. vacuum.

If the exhaust is fully condensed to A, cool condensate will be used as boiler feed.

We have then in the straight condensation cycle :—

Heat input	C − A = 1,198 − 129 =	1,069 Btu/lb.
Heat drop	C − D = 1,198 − 951 =	247 „
Cycle efficiency	$\frac{247 \times 100}{1,069}$ =	23·1 per cent.

The Compression-Condensation cycle is shown in Fig. 436 as E B C D.

Steam at C, 160 psi saturated, is expanded to D, 20-in. vacuum.

The exhaust is partly condensed to E, about 82 per cent. wet.

The steam-condensate mixture is compressed from E to boiler pressure and temperature B.

The state point has traced out C D E B—a Carnot rectangle.

Heat input	C − B = 1,198 − 344 =	854 Btu/lb.
Heat drop	C − D = 1,198 − 951 =	247 „
Compressor power	B − E = 344 − 313 =	31 „
Available net power	247 − 31 =	216 „
Cycle efficiency	$\frac{216 \times 100}{854}$ =	25·3 per cent.

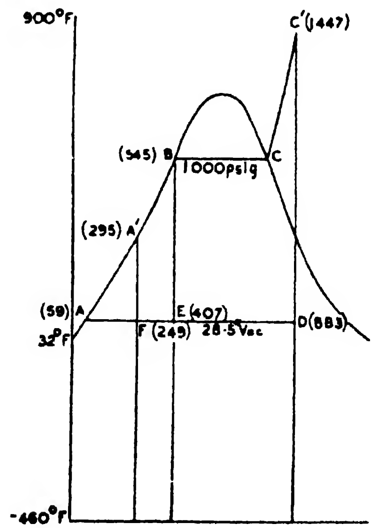

FIG. 437.

This shows a gain over straight condensation of 2·2 on 23·1 or 9·5 per cent. It is extremely unlikely that any such gain would be achieved in practice.

In order that the cycle efficiency be equal to that of straight condensation the available net power must be equal to

$$\frac{23 \cdot 1 \times 854}{100} = 197 \text{ Btu/lb.}$$

∴ The compressor power must equal

$$247 - 197 = 50 \quad ,,$$

giving an overall efficiency for the compressor and its drive of

$$\frac{31}{50} = 62 \text{ per cent.}$$

which is not very likely.

Fig. 437 shows the Compression-Condensation cycle applied to a high pressure power plant.

The regenerative cycle gives the following :—

Heat input	C′—A′	= 1,447 − 295 =	1,152 Btu/lb.
Heat drop	C′—D	= 1,447 − 883 =	564 ,,
Power forfeit by bleeds	A′— F	= 295 − 249 =	46 ,,
Net power available		564 − 46 =	518 ,,
Cycle efficiency		$\frac{518 \times 100}{1,152}$ =	45 per cent.

The compression condensation cycle gives :—

Heat input	C′— B	= 1,447 − 545 =	902 Btu/lb.
Heat drop	C′— D	= 1,447 − 883 =	564 ,,
Compressor power	B — E	= 545 − 407 =	138 ,,
Net power available		564 − 138 =	426 ,,
Cycle efficiency		$\frac{426 \times 100}{902}$ =	47·2 per cent.

The improvement theoretically obtained with compression-condensation over bleed heating is very small, only 5 per cent. There is little hope of the gain in efficiency not being swamped by compressor losses. The compressor would be a huge formidable machine. It must be very multistage, and must start as a blower taking a mixture of steam and water, and must end up as a centrifugal pump on water only.

The cycle has actually been tried on a small power set and was reported to show a very great improvement—much greater than was theoretically possible. It was also tried on a locomotive, but the lack of exhaust to atmosphere to produce draught completely damned it.

Gas Cycles.

All the steam cycles discussed in this book have been vapour-liquid cycles. There is no reason why the fluid should exist in the liquid phase at any part of

the cycle. Such a dry cycle is called a gas cycle and can show a very high theoretical efficiency. Air or some dense permanent gas would be more convenient than steam.

J. P. Joule (a brewer) and Robert Stirling (a parson) devised air cycles 100 years ago that had cycle efficiencies two or three times those of the steam cycles in contemporary use. A 50 horse-power Stirling engine was actually worked in a Scottish factory 100 years ago and gave the surprising figure of 2·7 lb. coal/BHP or, in modern parlance, 3·6 lb./kWh. Reference to Table XII will show how truly remarkable this was. These old gas cycles failed because the temperatures were too high for contemporary materials to withstand and the volume of the engine had to be huge.

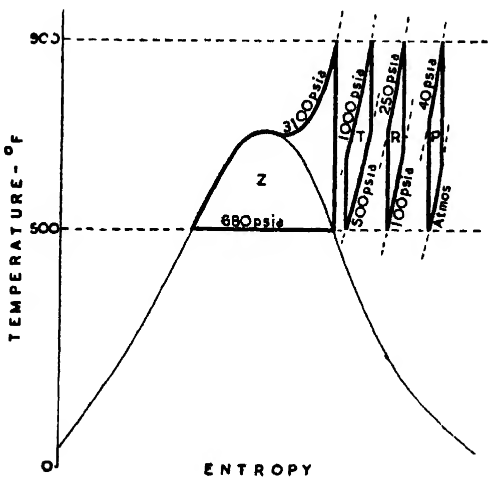

FIG. 438. GAS CYCLES USING STEAM

With the advent of the modern very efficient turbo-blowers and gas combustion turbines, it is quite likely that gas-cycle plants, using some suitable gas, may become serious competitors of steam.

As this book deals with steam and contains no data on other fluids, a comparison of a vapour-liquid steam cycle will be made with superheated steam gas cycles.

The gas cycles will be assumed to work thus :—Superheated steam at 900° F. will expand through a turbine to about double its volume. It will then reject heat by being cooled to 500° F. After cooling it will enter a turbo-compressor, driven by the turbine and be compressed back to its initial pressure. It will then pass through a fuel-fired superheater where it will be heated up again to 900° F. The cycle then repeats. Fastened to the end of the huge turbo-compressor is a relatively small generator. The bottom temperature of 500° F. has been chosen because several pressure drops between 900° F. and 500° F. can be compared without risk of getting into the wet region.

Fig. 438 is a temperature/entropy diagram showing three of these gas cycles, T, R and P, using superheated steam. The vapour-liquid steam cycle between the same temperatures is shown at Z.

These four cycles, and two not shown in Fig. 438, are tabulated below.

	INITIAL			EXHAUST		AFTER REJECT OR PROCESS		RECOMPRESSED TO		HEAT INPUT	POWER OUT	COMPR. POWER	NET POWER OUT	CYCLE EFFY.	HEAT PUT OUT
	PSI.A.	°F.	BTU A	PSI.A.	BTU B	°F.	BTU C	PSI.A.	BTU D	A—D = E	A—B = F	D—C = G	F—G = H	$\frac{100H}{E}$ %	B—C
T	1,000	900	1,448	500	1,358	500	1,232	1,000	1,298	150	90	66	24	16	126
S	500	900	1,467	250	1,373	500	1,264	500	1,340	127	94	76	18	14·2	109
R	250	900	1,474	100	1,353	500	1,279	250	1,386	88	121	107	14	15·9	74
Q	100	900	1,479	40	1,356	500	1,285	100	1,393	86	123	108	15	17·5	71
P	40	900	1,482	AT.	1,347	500	1,288	40	1,408	74	135	120	15	20·3	59
										A—C			A—B		
Z	3,100	900	1,357	680	1,203	500	489			868	154		154	17·8	714

It will be seen that the theoretical efficiency of these gas cycles is little different from that given by the corresponding vapour-liquid cycle. There are very great other differences however. Cycle P gives only one tenth of the power given by cycle Z per lb. of steam. For the same output the plant in cycle P must have a steam throughput of ten times that of cycle Z. The average specific volume of the steam in cycle P is 60 times that in cycle Z. So that for equal powers a plant working cycle P would have to pass several hundred times the volume of the steam in cycle Z. On the other hand cycle Z is working at the very high pressure of 3,100 psi whereas cycle P works between 40 psi.a. and atmospheric pressure.

The advantages of gas cycles are that much lower pressures and lower temperatures can be used. The principal disadvantage is the size of the machines. Most of the work done in the turbine is used to drive the compressor.

The object of discussing gas cycles here is simply to show that a steam power plant could be made to work without liquid water existing at any part of the cycle, and to give a brief explanation of the way gas cycles work, as they may be in extensive use in a few years' time. Steam is not very suitable for use in a gas cycle. The best fluid is a dense permanent gas.

804. PARALLELING WITH THE GRID. In many factories steam demand and power demand are out of balance. In a number of factories the power load is fairly constant, but the steam load is very heavy at the beginning of the day and rapidly tails off. In such factories a back pressure engine may only be able to meet a small part of the afternoon electrical load, though the morning steam demand could possibly have produced twice the power that the factory needed.

In such a case it is clear that the right way to run is in parallel with the grid. In the morning, current could be fed into the grid and in the afternoon current could be drawn from the grid.

There are a number of factories that operate thus, but they are a drop in the ocean compared to the number that ought to be so working. Some electricity undertakings do not look favourably on parallel running and have in the past raised all kinds of objections. These objections are possibly exaggerated as those works that do run in parallel do so quite successfully.

Once synchronisation is effected the machine governor goes out of action and the output is dependent on the throttle opening, which should be controlled by the process steam demand. The voltage is that of the mains supply and is not controlled by the factory generator excitation, which merely affects the power factor.

In the author's factory a turbo-generator of sufficient size to pass the whole of the process steam was installed in 1957. The generator runs in parallel with the grid and a profit of £600 a week is gained from exported current. The amount of steam passed through the turbine is controlled by the pressure in the accumulator. There seem to be no difficulties or disadvantages, nothing but benefits. Apart from the money gain the voltage and frequency are constant and the boiler load is more steady. The electrical load fluctuations are lost in the grid and process steam fluctuations are cushioned by the accumulator.

There are methods of running in parallel with the grid which do not call for synchronisation. The first is to use an induction generator. This is exactly like an induction motor. It is excited by the public supply and will deliver current into the public network if it is run fast enough. It simply leans against the public supply. In such a machine the governor should be controlled by the process demand, that is to say, the back pressure. Then all the steam that the process needs will go through the set and when the quantity is large the machine will speed up and export current. As the exciting current is drawn from the supply, unless there is a considerable current generated the power factor will be very low and it will almost certainly be necessary to provide a condenser to correct the power factor. There is such a set working in London and it is beautifully simple. It can be started and stopped without any synchronising gear or synchronising skill. The forfeit paid is a rather lower generator efficiency and the need for power factor correction.

The only other method that the author has heard of is the use of a synchronous induction generator. This is a machine exactly like an ordinary self-starting synchronous motor. The generator is provided with induction windings as well as synchronous poles. After being run up it can be switched on and it slips into synchronism automatically as soon as it approaches synchronous speed. The author has no information as to the practical working of such machines. They sound promising as there will be no synchronising hazards, yet the power factor and efficiency should be good.

There is one point on which the would-be power-seller must be warned. An electricity supply company will only pay a very meagre rate for exported current. If current is being bought for 1*d.* a unit, the supply company will probably offer something around a halfpenny. This is not deliberately "heads I win, tails you lose," but is an economic necessity. As a rule more than half of the cost of electricity is its distribution. Current exported by a factory only saves the supply company generating this current; it does not help the distribution problem except in a few isolated cases of overloaded feeders.

Quite a number of factories successfully share their electrical load between their own small generating plant and the public supply, by sectionalising their plant and providing two sets of bus bars. One set of bus bars is connected to their own generator and the other set is connected to the public supply. Change-over switches, which cannot contact both sets of bus bars simultaneously, are connected to each feeder so that flexibility of supply is assured. This arrangement is said to work well but it must result in more current being purchased than would have been imported with parallel running.

805. BOOKS. There are many books on boilers, heat engines, steam engines, turbines, thermodynamics, etc. There is a great lack of books on the use of steam from the factory operators' point of view.

The following list makes no pretence at being complete. It merely gives some of the books that the author has found helpful. Many of them are now out of print, and whether they will be reprinted within a reasonable time is uncertain. They are however available in many libraries.

When trying to learn a subject, one book is seldom enough. There are always places in every book where its author fails to get his explanation across. When trying to get a smattering of a new subject this author has always found it necessary to study at least two books, preferably three. What is foggy and muddled in one, is crystal clear in another. There are some books that read well but are little use for reference owing to the inadequacy of the Index. Other books must be read with a continually critical mind lest errors be accepted as facts. Most technical books in their first editions contain errors, and a first edition should always be read with this in mind. The searching for mistakes in a book is one of the best ways of really learning the subject.

Here is a brief list of books useful to the steam user :—

The 1939 *Callendar Steam Tables*—by G. S. Callendar and A. C. Egerton.
Edward Arnold & Co. 1944. 2nd Edition. 65 pages.

The Thermodynamic Properties of Steam—by Joseph H. Keenan and Frederick G. Keyes.
Chapman & Hall, Ltd. 1936. 1st Edition. 90 pages.

Of the two steam tables Keenan and Keyes is more conveniently arranged, Callendar is more modern.

The 1939 *Heat-Entropy Diagram for Steam*—3 colours. 38 in. by 33 in.
Edward Arnold.

This is a first-rate Mollier diagram of sufficient size to enable readings to be secured to 1 Btu.

The Efficient Use of Fuel—written for the Fuel Efficiency Committee by many people.

H.M. Stationery Office. 1944. 1st Edition. 800 pages.

This book fills a great want by collecting together in one cheap volume a huge amount of valuable information. Its value would be greatly enhanced by a proper Index.

Steam Engine Theory and Practice—by William Ripper. (Out of print.)

Longmans Green. 6th Edition. 1922. 500 pages.

A first-class book which never lets theory get too far away from practice. The mathematics are kept to a minimum.

Steam Power—by W. A. Dalby. (Out of print.)

Edward Arnold. 2nd Edition. 1920. 750 pages.

This is perhaps the best book that has ever been written on Steam Engines.

Heat Engines—by David Allan Low.

Longmans, Green & Co., Ltd. 1930. 1st Edition. 15th Imp.

This is one of the best books on Heat Engines as it is full of practical references. The illustrations are particularly good. It deals only with Heat Engines.

The Theory and Practice of Heat Engines—by D. A. Wrangham.

Cambridge University Press. 1st Edition. 1942. 750 pages.

This is a fine up-to-date book full of information, but also full of fairly advanced mathematics. It only deals with power plants and refrigerators.

The Theory and Practice of Heat Engines—by R. H. Grundy.

Longmans Green. 1st Edition. 1942. 720 pages.

An excellent up-to-date book. Deals only with heat engines.

Valves and Valve Gears. Vol. I—by F. D. Furman. (Out of print.)

Chapman & Hall, Ltd. 1923. 2nd Edition. 250 pages.

This book gives much information about the characteristics of various valve gears. Describes the various graphical methods for analysing valve performance.

Exhaust Steam Engineering—by Charles S. Darling. (Out of print.)

Chapman & Hall, Ltd. 1928. 1st Edition. 430 pages.

The title of this book is rather misleading. Half the book is concerned with turbines—by no means only the exhaust end. But there is a lot of useful information about back pressure and pass-out arrangements and about accumulators. It contains many mistakes.

Steam for Process and Industrial Heating—by Alexander H. Hayes.

Forum Publishing Company. 2nd Edition. 1942. 200 pages.

This is an excellent little book, but it contains more mistakes than a second edition should. It is very easy reading, contains hardly any formulae and very little arithmetic.

Steam Trapping and Air Venting—by L. G. Northcroft.

Hutchinson. 2nd Edition. 1945. 170 pages.

This is an excellent book, full of practical examples, well illustrated, racily written. It fills a great and long-felt want. It deserves a much better index.

Evaporating, Condensing and Cooling Apparatus—by E. Hausbrand. (Out of print.)

Ernest Benn. 5th Edition. 1933. 500 pages.

This is the standard text book on evaporator heat transfer. It is full of good stuff, but there is a lot of theory (especially on entrainment) that seems inadequately verified by practice.

Heat Transfer and Evaporation—by W. L. Badger. (Out of print.)

Chemical Catalog Company. 1st Edition. 1926. 300 pages.

This is a first-class book—possibly the best on the subject. Multiple effect evaporation is dealt with in detail. It is readable with not too much mathematics.

Evaporation—by Alfred L. Webre. (Out of print.)

Chemical Catalog Company. 1st Edition. 1926. 500 pages.

An interesting and readable book. All kinds of Multiple Effect Plants are analysed in detail. The book contains practically no advanced mathematics.

Heat Transmission—by William H. McAdams.

McGraw-Hill. 2nd Edition. 1942. 450 pages.

An advanced book in which are collected almost all the published heat transfer figures. Although it is the standard book on the subject, it is not suitable for ordinary, practical men, unless they are technically qualified.

Heating and Air Conditioning of Buildings—by Oscar Faber and J. R. Kell.

The Architectural Press. 1943. 2nd Edition. 580 pages.

This book contains a mass of practical information with very little theory. All types of heating are discussed and the tables and diagrams are numerous and useful.

Boiler Feed Water Treatment—by F. J. Matthews.

Hutchinson. 3rd Edition. 1946. 250 pages.

This book gives, in a simple easily read way, much information with the minimum of chemistry.

Water Treatment Manual—Babcock & Wilcox, Ltd.

(Undated.) 100 pages.

A practical book which describes a comprehensive water control.

Properties of Ordinary Water Substance—by N. Ernest Dorsey.

Reinhold Publishing Corporation. 1940. 1st Edition. 670 pages.

This is a really high-brow book and very difficult to use for ordinary work. It is a collection, in some 280 tables, of all the authoritative published data on Ice—Water—Steam. The units used are all International and are very tiresome. Scattered throughout the book is an astonishing amount of useful miscellaneous information.

Regression Analysis of Production Costs and Factory Operations—by Philip Lyle.

Oliver & Boyd. 2nd Edition. 1946. 200 pages.

This book discusses Regression Analysis in detail and has an excellent section on fitting simple empirical equations to all kinds of data. Although all the mathematics are quite elementary, the book requires concentration from the reader because it is very concise.

806. STEAM TABLES. H. V. Regnault, about 100 years ago, made the first systematic measurements of the heat properties of water and steam. Prof. H. L. Callendar published his epoch-making work on the properties of steam in 1900. Since that time much research has been done by many eminent workers all over the world. By 1930 satisfactory practical agreement was reached for most of the properties over a very wide range. For the ordinary factory technologist any steam table published since 1930 will be quite accurate enough for all ordinary purposes.

The tables in most common use in Britain are :—

Keenan, 1930.
Callendar, 1931.
Keenan and Keyes, 1936.
Callendar, 1939.

There are some small inconsistencies within the Callendar 1939 tables, but they are of no importance.

Steam tables seem to be constructed for the convenience of a handful of turbine designers. They are either based on absolute pressures or on temperatures. The factory operator seldom knows his temperature and generally talks only of gauge pressure, so that to use a steam table he almost always has to indulge in tiresome interpolation. The Saturation and Superheat tables following have been based on gauge pressures. It is hoped that the intervals and range are such as to reduce interpolation to a minimum without making the tables unduly cumbersome.

These steam tables have been constructed by interpolation from the plotted and smoothed values in the Callendar 1939 table.

Steam tables contain the whole story of steam, and this story has been elaborated in this book.

TABLES

TABLE I. THE PROPERTIES OF SATURATED STEAM AND WATER*

PRESSURE		TEMPERATURE °F.	HEAT BTU/LB.				VOLUME CU. FT./LB.		ENTROPY BTU/°F.ABS./LB.		
VAC. IN. HG.	PSI ABS.		SENSIBLE	LATENT	TOTAL	GIBBS	LIQUID	VAPOUR	LIQUID	EVAP.	TOTAL
29·5	·245	58·8	26·8	1060·5	1087·3	0·6	·0160	1257	·053	2·045	2·098
29·25	·367	70·2	38·3	1053·8	1092·1	1·3	·0161	860	·075	1·989	2·064
29	·490	78·9	47·0	1049·1	1096·1	2·0	·0161	656	·091	1·948	2·039
28·75	·612	85·8	53·8	1045·2	1099·0	2·7	·0161	532	·104	1·916	2·020
28·5	·735	91·6	59·6	1041·9	1101·5	3·4	·0161	447	·114	1·890	2·004
28·25	·857	96·6	64·6	1039·3	1103·9	4·0	·0161	387	·123	1·868	1·991
28	·979	101·0	69·0	1036·7	1105·7	4·6	·0161	341	·131	1·849	1·980
27·75	1·102	105·0	72·9	1034·6	1107·5	5·1	·0162	305	·138	1·832	1·970
27·5	1·224	108·6	76·6	1032·4	1109·0	5·6	·0162	276	·145	1·816	1·961
27·25	1·347	111·8	79·9	1030·6	1110·5	6·1	·0162	252	·150	1·803	1·953
27	1·469	114·9	82·9	1028·9	1111·8	6·6	·0162	233	·156	1·790	1·946
26·5	1·714	120·5	88·4	1025·7	1114·1	7·5	·0162	201	·165	1·768	1·933
26	1·958	125·3	93·2	1022·8	1116·0	8·3	·0162	177	·174	1·748	1·922
25·5	2·203	129·7	97·6	1020·3	1117·9	9·0	·0163	159	·181	1·732	1·913
25	2·448	133·6	101·5	1018·1	1119·6	9·8	·0163	144	·188	1·716	1·904
24·5	2·693	137·3	105·2	1015·9	1121·1	10·5	·0163	132	·194	1·702	1·896
24	2·938	140·7	108·7	1013·9	1122·6	11·2	·0163	121	·200	1·689	1·889
23·5	3·183	143·8	111·8	1012·1	1123·9	11·9	·0163	112	·205	1·677	1·882
23	3·428	146·8	114·7	1010·3	1125·0	12·5	·0163	105	·210	1·666	1·876
22·5	3·672	149·5	117·4	1008·8	1126·2	13·1	·0163	98·4	·214	1·657	1·871
22	3·917	152·1	120·0	1007·2	1127·2	13·7	·0164	92·6	·218	1·647	1·865
21·5	4·162	154·6	122·5	1005·8	1128·3	14·2	·0164	87·5	·222	1·638	1·860
21	4·407	157·0	124·9	1004·2	1129·1	14·7	·0164	82·9	·226	1·629	1·855
20	4·896	161·4	129·3	1001·7	1131·0	15·8	·0164	75·1	·233	1·614	1·847
19	5·386	165·4	133·3	999·4	1132·7	16·8	·0164	68·7	·240	1·599	1·839
18	5·876	169·2	137·1	997·1	1134·2	17·7	·0164	63·3	·246	1·586	1·832
17	6·365	172·7	140·6	995·0	1135·6	18·5	·0165	58·8	·252	1·573	1·825
16	6·855	175·9	143·8	993·1	1136·9	19·3	·0165	54·8	·257	1·562	1·819
15	7·344	178·9	146·8	991·3	1138·1	20·0	·0165	51·4	·261	1·552	1·813
14	7·834	181·9	149·8	989·4	1139·2	20·8	·0165	48·4	·266	1·542	1·808
13	8·324	184·7	152·7	987·7	1140·4	21·6	·0165	45·7	·270	1·533	1·803
12	8·813	187·3	155·3	986·1	1141·4	22·3	·0166	43·3	·274	1·524	1·798
11	9·303	189·8	157·8	984·7	1142·5	23·0	·0166	41·2	·278	1·516	1·794
10	9·793	192·2	160·2	983·2	1143·4	23·7	·0166	39·2	·282	1·508	1·790
9	10·28	194·5	162·5	981·8	1144·3	24·3	·0166	37·5	·286	1·500	1·786
8	10·77	196·8	164·8	980·4	1145·2	25·0	·0166	35·9	·289	1·493	1·782
7	11·26	198·9	167·0	979·1	1146·1	25·6	·0166	34·4	·292	1·487	1·779
6	11·75	201·0	169·1	977·7	1146·8	26·2	·0166	33·1	·295	1·480	1·775
5	12·24	203·0	171·0	976·5	1147·5	26·8	·0167	31·9	·298	1·474	1·772
4	12·73	204·9	172·9	975·3	1148·2	27·4	·0167	30·7	·301	1·468	1·769
3	13·22	206·8	174·8	974·0	1148·8	27·9	·0167	29·7	·304	1·462	1·766
2	13·71	208·6	176·6	972·9	1149·5	28·5	·0167	28·7	·307	1·456	1·763
1	14·20	210·3	178·4	971·7	1150·1	29·0	·0167	27·7	·310	1·450	1·760

* The Tables giving the properties of saturated steam and water and the properties of superheated steam are based on the values given in the 1939 Callendar Steam Tables by permission of Messrs. Edward Arnold and Co.

TABLE I. THE PROPERTIES OF SATURATED STEAM AND WATER *continued*

PRESSURE		TEMPERATURE °F.	HEAT BTU/LB.				VOLUME CU. FT./LB.		ENTROPY BTU/°F.ABS./LB.		
PSI GAUGE	PSI ABS.		SENSIBLE	LATENT	TOTAL	GIBBS	LIQUID	VAPOUR	LIQUID	EVAP.	TOTAL
0	14·69	212	180·2	970·6	1150·8	29·9	·0167	26·8	·312	1·445	1·757
1	15·7	215·4	183·6	968·4	1152·0	31·0	·0167	25·2	·317	1·435	1·752
2	16·7	218·5	186·8	966·4	1153·2	32·1	·0168	23·8	·322	1·425	1·747
3	17·7	221·5	189·8	964·5	1154·3	33·0	·0168	22·5	·326	1·416	1·742
4	18·7	224·5	192·7	962·6	1155·3	34·0	·0168	21·4	·331	1·407	1·738
5	19·7	227·4	195·5	960·8	1156·3	34·9	·0168	20·4	·335	1·399	1·734
6	20·7	230·0	198·1	959·2	1157·3	35·8	·0168	19·4	·339	1·391	1·730
7	21·7	232·4	200·6	957·6	1158·2	36·7	·0169	18·6	·342	1·384	1·726
8	22·7	234·8	203·1	956·0	1159·1	37·5	·0169	17·9	·346	1·376	1·722
9	23·7	237·1	205·5	954·5	1160·0	38·3	·0169	17·2	·349	1·370	1·719
10	24·7	239·4	207·9	952·9	1160·8	39·1	·0169	16·5	·352	1·363	1·716
11	25·7	241·6	210·1	951·5	1161·6	39·8	·0169	15·9	·356	1·356	1·712
12	26·7	243·7	212·3	950·1	1162·3	40·6	·0170	15·3	·359	1·350	1·709
13	27·7	245·8	214·4	948·6	1163·0	41·3	·0170	14·8	·362	1·344	1·706
14	28·7	247·9	216·4	947·3	1163·7	42·0	·0170	14·3	·365	1·338	1·703
15	29·7	249·8	218·4	946·0	1164·4	42·7	·0170	13·9	·367	1·333	1·700
16	30·7	251·7	220·3	944·8	1165·1	43·4	·0170	13·4	·370	1·328	1·698
17	31·7	253·6	222·2	943·5	1165·7	44·1	·0170	13·0	·373	1·323	1·696
18	32·7	255·4	224·0	942·4	1166·4	44·8	·0170	12·7	·375	1·318	1·693
19	33·7	257·2	225·8	941·2	1167·0	45·4	·0171	12·3	·378	1·313	1·691
20	34·7	258·8	227·5	940·1	1167·6	46·1	·0171	12·0	·380	1·308	1·688
21	35·7	260·5	229·2	939·0	1168·2	46·8	·0171	11·7	·382	1·304	1·686
22	36·7	262·3	230·9	937·8	1168·7	47·4	·0171	11·4	·385	1·299	1·684
23	37·7	263·7	232·6	936·7	1169·3	48·1	·0171	11·1	·387	1·295	1·682
24	38·7	265·3	234·2	935·8	1169·8	48·7	·0171	10·8	·389	1·291	1·680
25	39·7	266·8	235·8	934·6	1170·4	49·3	·0171	10·6	·391	1·287	1·678
26	40·7	268·3	237·3	933·5	1170·8	49·8	·0172	10·3	·393	1·283	1·676
27	41·7	269·8	238·7	932·6	1171·3	50·3	·0172	10·1	·395	1·279	1·674
28	42·7	271·4	240·2	931·6	1171·8	50·8	·0172	9·87	·397	1·275	1·672
29	43·7	272·6	241·6	930·6	1172·2	51·3	·0172	9·66	·399	1·271	1·670
30	44·7	274·0	243·0	929·7	1172·7	51·8	·0172	9·46	·401	1·267	1·668
31	45·7	275·4	244·4	928·7	1173·1	52·3	·0172	9·27	·403	1·263	1·666
32	46·7	276·7	245·9	927·6	1173·5	52·8	·0172	9·08	·405	1·260	1·665
33	47·7	278·1	247·2	926·7	1173·9	53·3	·0172	8·90	·407	1·256	1·663
34	48·7	279·4	248·5	925·8	1174·3	53·8	·0173	8·73	·409	1·252	1·661
35	49·7	280·7	249·8	924·9	1174·7	54·3	·0173	8·56	·410	1·249	1·659
36	50·7	281·9	251·1	924·0	1175·1	54·8	·0173	8·40	·412	1·246	1·658
37	51·7	283·2	252·4	923·1	1175·5	55·3	·0173	8·25	·414	1·242	1·656
38	52·7	284·4	253·7	922·1	1175·8	55·8	·0173	8·11	·416	1·239	1·655
39	53·7	285·6	254·9	921·3	1176·2	56·4	·0173	7·97	·417	1·236	1·653
40	54·7	286·7	256·1	920·4	1176·5	56·9	·0173	7·83	·419	1·233	1·652
41	55·7	287·9	257·3	919·5	1176·8	57·4	·0173	7·70	·420	1·230	1·650
42	56·7	289·0	258·5	918·6	1177·1	57·9	·0174	7·57	·422	1·227	1·649
43	57·7	290·1	259·6	917·9	1177·5	58·4	·0174	7·45	·424	1·224	1·648
44	58·7	291·3	260·8	917·0	1177·8	58·9	·0174	7·33	·425	1·221	1·646
45	59·7	292·4	261·9	916·2	1178·1	59·3	·0174	7·22	·427	1·218	1·645
46	60·7	293·5	263·0	915·4	1178·4	59·8	·0174	7·10	·428	1·216	1·644
47	61·7	294·5	264·1	914·6	1178·7	60·4	·0174	6·99	·430	1·212	1·642
48	62·7	295·6	265·2	913·8	1179·0	60·9	·0174	6·89	·431	1·210	1·641
49	63·7	296·6	266·3	913·0	1179·3	61·4	·0174	6·78	·433	1·207	1·640
50	64·7	297·7	267·4	912·2	1179·6	61·9	·0174	6·68	·434	1·204	1·638
51	65·7	298·7	268·4	911·5	1179·9	62·4	·0174	6·59	·435	1·202	1·637
52	66·7	299·7	269·4	910·7	1180·1	62·9	·0175	6·50	·437	1·199	1·636
53	67·7	300·7	270·4	910·0	1180·4	63·4	·0175	6·41	·438	1·197	1·635
54	68·7	301·7	271·5	909·2	1180·7	63·8	·0175	6·32	·439	1·195	1·634
55	69·7	302·7	272·5	908·5	1181·0	64·2	·0175	6·24	·441	1·192	1·633

TABLE I. THE PROPERTIES OF SATURATED STEAM AND WATER *continued*

PRESSURE		TEMPERATURE °F.	HEAT BTU/LB.				VOLUME CU. FT./LB.		ENTROPY BTU/°F.ABS./LB.		
PSI GAUGE	PSI ABS.		SENSIBLE	LATENT	TOTAL	GIBBS	LIQUID	VAPOUR	LIQUID	EVAP.	TOTAL
56	70·7	303·6	273·5	907·8	1181·3	64·6	·0175	6·16	·442	1·189	1·631
57	71·7	304·6	274·4	907·2	1181·6	65·0	·0175	6·08	·443	1·187	1·630
58	72·7	305·5	275·3	906·5	1181·8	65·4	·0175	6·00	·444	1·185	1·629
59	73·7	306·5	276·2	905·9	1182·1	65·8	·0175	5·92	·446	1·182	1·628
60	74·7	307·4	277·1	905·3	1182·4	66·2	·0175	5·84	·447	1·180	1·627
61	75·7	308·3	278·0	904·7	1182·7	66·6	·0176	5·77	·448	1·178	1·626
62	76·7	309·2	279·0	904·0	1183·0	67·0	·0176	5·70	·449	1·176	1·625
63	77·7	310·0	280·0	903·2	1183·2	67·4	·0176	5·63	·451	1·173	1·624
64	78·7	310·9	280·9	902·6	1183·5	67·7	·0176	5·56	·452	1·171	1·623
65	79·7	311·8	281·8	901·9	1183·7	68·1	·0176	5·50	·453	1·169	1·622
66	80·7	312·7	282·8	901·2	1184·0	68·4	·0176	5·43	·454	1·167	1·621
67	81·7	313·5	283·7	900·5	1184·2	68·8	·0176	5·37	·455	1·165	1·620
68	82·7	314·3	284·5	900·0	1184·5	69·1	·0176	5·31	·456	1·163	1·619
69	83·7	315·2	285·3	899·4	1184·7	69·5	·0176	5·25	·458	1·160	1·618
70	84·7	316·0	286·2	898·8	1185·0	69·8	·0176	5·19	·459	1·158	1·617
71	85·7	316·9	287·2	898·0	1185·2	70·1	·0176	5·13	·460	1·156	1·616
72	86·7	317·7	288·0	897·5	1185·5	70·5	·0176	5·08	·461	1·154	1·615
73	87·7	318·5	288·7	897·0	1185·7	70·8	·0177	5·02	·462	1·152	1·614
74	88·7	319·3	289·4	896·5	1185·9	71·2	·0177	4·97	·463	1·150	1·613
75	89·7	320·1	290·3	895·8	1186·1	71·5	·0177	4·92	·464	1·148	1·612
76	90·7	320·9	291·2	895·1	1186·3	71·9	·0177	4·87	·465	1·146	1·611
77	91·7	321·7	292·0	894·5	1186·5	72·2	·0177	4·82	·466	1·145	1·611
78	92·7	322·4	292·9	893·9	1186·8	72·5	·0177	4·77	·467	1·143	1·610
79	93·7	323·2	293·7	893·3	1187·0	72·8	·0177	4·72	·468	1·141	1·609
80	94·7	323·9	294·5	892·7	1187·2	73·1	·0177	4·67	·469	1·139	1·608
81	95·7	324·7	295·3	892·1	1187·4	73·4	·0177	4·63	·470	1·137	1·607
82	96·7	325·5	296·1	891·5	1187·6	73·8	·0177	4·58	·471	1·135	1·606
83	97·7	326·2	296·8	890·9	1187·7	74·1	·0177	4·53	·472	1·133	1·605
84	98·7	326·9	297·6	890·3	1187·9	74·5	·0177	4·49	·473	1·132	1·605
85	99·7	327·7	298·3	889·8	1188·1	74·8	·0177	4·45	·474	1·130	1·604
86	100·7	328·4	299·1	889·2	1188·3	75·1	·0178	4·41	·475	1·128	1·603
87	101·7	329·1	299·8	888·7	1188·5	75·5	·0178	4·37	·476	1·126	1·602
88	102·7	329·9	300·6	888·1	1188·7	75·8	·0178	4·33	·477	1·124	1·601
89	103·7	330·5	301·3	887·5	1188·8	76·2	·0178	4·29	·478	1·123	1·601
90	104·7	331·2	302·1	887·0	1189·1	76·5	·0178	4·25	·479	1·121	1·600
91	105·7	331·9	302·8	886·4	1189·2	76·9	·0178	4·21	·480	1·119	1·599
92	106·7	332·6	303·5	885·8	1189·3	77·2	·0178	4·17	·480	1·118	1·598
93	107·7	333·3	304·2	885·3	1189·5	77·6	·0178	4·14	·481	1·117	1·598
94	108·7	333·9	304·9	884·8	1189·7	77·9	·0178	4·10	·482	1·115	1·597
95	109·7	334·6	305·6	884·2	1189·8	78·2	·0178	4·07	·483	1·113	1·596
96	110·7	335·3	306·3	883·7	1190·0	78·6	·0178	4·03	·484	1·111	1·595
97	111·7	335·9	307·0	883·2	1190·2	78·9	·0178	4·00	·485	1·110	1·595
98	112·7	336·6	307·7	882·6	1190·3	79·3	·0178	3·96	·486	1·108	1·594
99	113·7	337·3	308·3	882·2	1190·5	79·6	·0178	3·93	·486	1·107	1·593
100	114·7	337·9	309·0	881·6	1190·6	79·9	·0178	3·90	·487	1·105	1·592
102	116·7	339·2	310·3	880·6	1190·9	80·5	·0178	3·83	·489	1·102	1·591
104	118·7	340·5	311·6	879·6	1191·2	81·2	·0179	3·77	·491	1·099	1·590
106	120·7	341·7	313·0	878·5	1191·5	81·8	·0179	3·71	·492	1·096	1·588
108	122·7	343·0	314·3	877·5	1191·8	82·4	·0179	3·65	·494	1·093	1·587
110	124·7	344·2	315·5	876·5	1192·0	83·0	·0179	3·60	·495	1·091	1·586
112	126·7	345·4	316·8	875·5	1192·3	83·6	·0179	3·54	·497	1·087	1·584
114	128·7	346·5	318·0	874·5	1192·5	84·2	·0179	3·49	·499	1·084	1·583
116	130·7	347·7	319·3	873·5	1192·8	84·8	·0179	3·44	·500	1·082	1·582
118	132·7	348·9	320·5	872·5	1193·0	85·4	·0180	3·39	·502	1·079	1·581
120	134·7	350·1	321·8	871·5	1193·3	86·0	·0180	3·34	·503	1·076	1·579
122	136·7	351·2	322·9	870·8	1193·6	86·6	·0180	3·30	·505	1·073	1·578
124	138·7	352·3	324·1	869·8	1193·9	87·2	·0180	3·25	·506	1·071	1·577
126	140·7	353·4	325·2	868·9	1194·1	87·7	·0180	3·21	·508	1·068	1·576
128	142·7	354·5	326·4	867·9	1194·3	88·3	·0180	3·16	·509	1·066	1·575
130	144·7	355·6	327·6	866·9	1194·5	88·9	·0180	3·12	·510	1·063	1·573

TABLE I. THE PROPERTIES OF SATURATED STEAM AND WATER *continued*

PRESSURE		TEMPERATURE °F.	HEAT BTU/LB.			VOLUME CU. FT./LB.		ENTROPY BTU/°F.ABS./LB.		
PSI GAUGE	PSI ABS.		SENSIBLE	LATENT	TOTAL	LIQUID	VAPOUR	LIQUID	EVAP.	TOTAL
132	146·7	356·7	328·8	865·9	1194·7	·0180	3·08	·512	1·060	1·572
134	148·7	357·8	330·0	865·0	1195·0	·0180	3·04	·513	1·058	1·571
136	150·7	358·8	331·1	864·1	1195·2	·0181	3·00	·514	1·056	1·570
138	152·7	359·9	332·2	863·3	1195·5	·0181	2·96	·516	1·053	1·569
140	154·7	360·9	333·2	862·5	1195·7	·0181	2·93	·517	1·051	1·568
142	156·7	361·9	334·3	861·6	1195·9	·0181	2·89	·518	1·049	1·567
144	158·7	362·9	335·4	860·7	1196·1	·0181	2·86	·520	1·046	1·566
146	160·7	364·0	336·4	859·9	1196·3	·0181	2·82	·521	1·044	1·565
148	162·7	365·0	337·5	858·9	1196·4	·0182	2·79	·522	1·042	1·564
150	164·7	365·9	338·6	858·0	1196·6	·0182	2·76	·523	1·040	1·563
155	169·7	368·3	341·1	856·0	1197·1	·0182	2·68	·526	1·034	1·560
160	174·7	370·7	343·6	853·9	1197·5	·0182	2·61	·529	1·029	1·558
165	179·7	372·9	346·1	851·8	1197·9	·0183	2·54	·532	1·024	1·556
170	184·7	375·2	348·5	849·8	1198·3	·0183	2·48	·535	1·018	1·553
175	189·7	377·5	350·9	847·9	1198·8	·0183	2·41	·538	1·013	1·551
180	194·7	379·6	353·2	845·9	1199·1	·0184	2·35	·540	1·009	1·549
185	199·7	381·6	355·4	844·1	1199·5	·0184	2·30	·543	1·004	1·547
190	204·7	383·7	357·6	842·2	1199·8	·0184	2·24	·546	·999	1·545
195	209·7	385·7	359·9	840·2	1200·1	·0184	2·18	·548	·995	1·543
200	214·7	387·7	362·0	838·4	1200·4	·0185	2·14	·551	·990	1·541
210	224·7	391·7	366·2	834·8	1201·0	·0185	2·04	·556	·981	1·537
220	234·7	395·5	370·3	831·2	1201·5	·0186	1·96	·561	·972	1·533
230	244·7	399·1	374·2	827·8	1202·0	·0186	1·88	·565	·964	1·529
240	254·7	402·7	378·0	824·5	1202·5	·0186	1·81	·570	·956	1·526
250	264·7	406·1	381·7	821·2	1202·9	·0187	1·74	·574	·947	1·523
260	274·7	409·3	385·3	817·9	1203·2	·0187	1·68	·578	·941	1·519
270	284·7	412·5	388·8	814·8	1203·6	·0188	1·62	·582	·934	1·516
280	294·7	415·8	392·3	811·6	1203·9	·0188	1·57	·586	·927	1·513
290	304·7	418·8	395·7	808·5	1204·2	·0189	1·52	·590	·920	1·510
300	314·7	421·7	398·9	805·5	1204·4	·0189	1·47	·593	·914	1·507
310	324·7	424·7	402·1	802·6	1204·7	·0190	1·43	·597	·908	1·505
320	334·7	427·5	405·2	799·7	1204·9	·0191	1·39	·601	·901	1·502
330	344·7	430·3	408·3	796·7	1205·0	·0191	1·35	·604	·895	1·499
340	354·7	433·0	411·3	793·8	1205·1	·0191	1·31	·607	·890	1·497
350	364·7	435·7	414·3	791·0	1205·3	·0192	1·27	·611	·883	1·494
360	374·7	438·3	417·2	788·2	1205·4	·0192	1·24	·614	·878	1·492
370	384·7	440·8	420·0	785·4	1205·4	·0193	1·21	·617	·872	1·489
380	394·7	443·3	422·8	782·7	1205·5	·0193	1·18	·620	·867	1·487
390	404·7	445·7	425·6	779·9	1205·5	·0194	1·15	·623	·862	1·485
400	414·7	448·1	428·2	777·4	1205·6	·0194	1·12	·626	·856	1·482
410	424·7	450·5	430·8	774·8	1205·6	·0195	1·09	·629	·851	1·480
420	434·7	452·8	433·4	772·2	1205·6	·0195	1·07	·632	·846	1·478
430	444·7	455·1	436·0	769·6	1205·6	·0195	1·04	·635	·841	1·476
440	454·7	457·3	438·5	767·1	1205·6	·0196	1·02	·637	·837	1·474
450	464·7	459·5	441·0	764·5	1205·5	·0196	1·00	·640	·832	1·472
460	474·7	461·7	443·4	762·1	1205·5	·0196	·979	·643	·827	1·470
470	484·7	463·8	445·9	759·5	1205·4	·0197	·959	·645	·823	1·468
480	494·7	465·9	448·3	757·1	1205·4	·0197	·939	·648	·818	1·466
490	504·7	467·9	450·6	754·7	1205·3	·0198	·920	·650	·814	1·464
500	514·7	470·0	453·0	752·3	1205·3	·0198	·902	·653	·809	1·462
510	524·7	472·0	455·3	749·9	1205·2	·0199	·885	·655	·805	1·460
520	534·7	474·0	457·6	747·5	1205·1	·0199	·868	·657	·801	1·458
530	544·7	475·9	459·8	745·2	1205·0	·0199	·852	·660	·796	1·456
540	554·7	477·8	462·0	742·8	1204·8	·0200	·835	·662	·792	1·454
550	564·7	479·7	464·2	740·5	1204·7	·0200	·820	·664	·789	1·453

TABLE I. THE PROPERTIES OF SATURATED STEAM AND WATER *continued*

PRESSURE		TEMPERATURE °F.	HEAT BTU/LB.			VOLUME CU. FT./LB.		ENTROPY BTU/°F.ABS./LB.		
PSI GAUGE	PSI ABS.		SENSIBLE	LATENT	TOTAL	LIQUID	VAPOUR	LIQUID	EVAP.	TOTAL
560	574·7	481·6	466·4	738·1	1204·5	·0200	·805	·667	·784	1·451
570	584·7	483·4	468·6	735·8	1204·4	·0201	·791	·669	·780	1·449
580	594·7	485·2	470·7	733·5	1204·2	·0201	·776	·671	·776	1·447
590	604·7	487·0	472·8	731·3	1204·1	·0201	·763	·673	·773	1·446
600	614·7	488·8	474·8	729·1	1203·9	·0202	·750	·676	·768	1·444
610	624·7	490·5	476·9	726·8	1203·7	·0202	·738	·678	·764	1·442
620	634·7	492·3	479·0	724·5	1203·5	·0203	·726	·680	·761	1·441
630	644·7	494·0	481·0	722·3	1203·3	·0203	·714	·682	·757	1·439
640	654·7	495·7	483·0	720·1	1203·1	·0203	·703	·684	·754	1·438
650	664·7	497·3	484·9	718·0	1202·9	·0204	·692	·686	·750	1·436
660	674·7	499·0	486·9	715·8	1202·7	·0204	·681	·688	·747	1·435
670	684·7	500·6	488·8	713·7	1202·5	·0204	·670	·690	·743	1·433
680	694·7	502·2	490·7	711·5	1202·2	·0205	·660	·692	·740	1·432
690	704·7	503·9	492·6	709·4	1202·0	·0205	·650	·694	·736	1·430
700	714·7	505·4	494·4	707·4	1201·8	·0206	·641	·696	·733	1·429
710	724·7	507·0	496·3	705·2	1201·5	·0206	·632	·697	·730	1·427
720	734·7	508·5	498·2	703·1	1201·3	·0206	·623	·699	·727	1·426
730	744·7	510·0	500·0	701·0	1201·0	·0207	·614	·701	·723	1·424
740	754·7	511·5	501·9	698·9	1200·8	·0207	·605	·703	·720	1·423
750	764·7	513·0	503·8	696·7	1200·5	·0208	·596	·705	·716	1·421
760	774·7	514·5	505·5	694·7	1200·2	·0208	·588	·707	·713	1·420
770	784·7	516·0	507·2	692·8	1200·0	·0208	·580	·708	·711	1·419
780	794·7	517·5	509·0	690·7	1199·7	·0209	·572	·710	·707	1·417
790	804·7	518·9	510·8	688·6	1199·4	·0209	·564	·712	·704	1·416
800	814·7	520·3	512·5	686·6	1199·1	·0209	·557	·714	·700	1·414
850	864·7	526·9	521·0	676·5	1197·5	·0211	·522	·722	·686	1·408
900	914·7	533·9	529·2	666·7	1195·9	·0213	·490	·730	·671	1·401
950	964·7	540·3	537·1	656·9	1194·0	·0215	·462	·738	·657	1·395
1000	1014·7	546·4	544·8	647·2	1192·0	·0217	·437	·745	·644	1·389
1050	1064·7	552·3	552·3	637·8	1190·0	·0218	·414	·753	·630	1·383
1100	1114·7	557·9	559·6	628·3	1187·9	·0220	·394	·760	·617	1·377
1150	1164·7	563·4	566·7	619·0	1185·7	·0222	·375	·766	·606	1·372
1200	1214·7	568·8	573·8	609·6	1183·4	·0223	·357	·773	·593	1·366
1250	1264·7	573·9	580·8	600·2	1181·0	·0225	·341	·780	·581	1·361
1300	1314·7	578·9	587·6	590 9	1178·5	·0227	·325	·786	·569	1·355
1350	1364·7	583·7	594·2	581·8	1176·0	·0229	·311	·792	·558	1·350
1400	1414·7	588·4	600·7	572·6	1173·3	·0231	·298	·798	·547	1·345
1450	1464·7	593·0	607·2	563·3	1170·5	·0233	·285	·804	·536	1·340
1500	1514·7	597·5	613·6	554·2	1167·8	·0235	·274	810	·525	1·335
1550	1564·7	601·8	619·8	545·2	1165·0	·0237	·264	816	·514	1·330
1600	1614·7	606·1	626·0	536·0	1162·0	·0239	·254	·821	·503	1·324
1700	1714·7	614·3	638·1	517·7	1155·8	·0243	·234	·832	·482	1·314
1800	1814·7	622·1	650·0	499·0	1149·1	·0248	·215	·843	·461	1·304
1900	1914·7	629·6	661·8	480·4	1142·2	·0253	·200	·853	·441	1·294
2000	2014·7	636·8	673·5	461·5	1135·0	·0258	·186	·863	·421	1·284
2100	2114·7	643·7	685·1	442·0	1127·1	·0263	·173	·874	·400	1·274
2200	2214·7	650·3	696·8	422·1	1118·9	·0268	·161	·884	·380	1·264
2300	2314·7	656·8	708·5	401·5	1110·0	·0274	·150	·894	·360	1·253
2400	2414·7	663·0	720·3	380·2	1100·5	·0281	·139	·904	·338	1·242
2500	2514·7	669·0	732 5	357·6	1090·1	·0289	·129	·914	·317	1·231
2600	2614·7	674·8	744·9	333·7	1078·6	·0297	·120	·925	·296	1·220
2700	2714·7	680·4	758·2	308·0	1066·2	·0306	·111	·935	·272	1·207
2800	2814·7	685·8	772·3	279·2	1051·5	·0318	·102	·947	·244	1·191
2900	2914·7	691·0	787·5	246·9	1034·4	·0332	·0936	·960	·215	1·175
3000	3014·7	696·1	805·3	207·4	1012·7	·0349	·0847	·975	·180	1·155
3193*	3208*	705·6*	896·0*	0	896.0*	·0489*	·0489*	1·052*	0	1·052*

* The values at the critical point were obtained by extrapolation.

TABLE II. THE PROPERTIES OF SUPERHEATED STEAM

VOLUME : CU. FT. PER LB.
TOTAL HEAT : BTU PER LB.
ENTROPY : BTU PER °F. ABS. PER LB.
} AT TEMPERATURES OF :—

PRESSURE: VACUUM IN. HG.	PRESSURE: PSI ABS. (SAT. TEMP.)	Sat.	110°F.	120°F.	130°F.	140°F.	150°F.	160°F.	170°F.	180°F.	190°F.	200°F.	220°F.	240°F.	260°F.	280°F.	300°F.	350°F.	400°F.
28	·979 (101)	341 1105·7 1·980	346 1110·3 1·987	352 1115·1 1·995	358 1119·6 2·003	364 1124·1 2·011	370 1128·6 2·018	376 1133·1 2·025	382 1137·6 2·033	388 1142·2 2·040	394 1146·6 2·046	400 1151·2 2·053	413 1160·4 2·068	425 1169·5 2·081	437 1178·5 2·094	450 1187·6 2·107	462 1196·7 2·119	492 1219·7 2·149	522 1242·4 2·176
27·5	1·224 (108·6)	276 1109·0 1·961	— — —	281 1114·9 1·972	286 1119·4 1·980	291 1124·0 1·988	296 1128·4 1·995	301 1133·0 2·003	306 1137·5 2·010	311 1142·1 2·017	315 1146·5 2·024	320 1151·1 2·031	330 1160·2 2·045	340 1169·2 2·058	350 1178·5 2·072	359 1187·5 2·085	369 1196·7 2·097	393 1219·7 2·126	418 1242·3 2·154
27	1·469 (114·9)	233 1111·8 1·946	— — —	235 1114·8 1·951	239 1119·3 1·959	243 1123·8 1·967	247 1128·3 1·974	251 1132·8 1·982	255 1137·3 1·990	259 1142·0 1·997	263 1146·4 2·005	267 1151·0 2·011	275 1160·1 2·025	283 1169·1 2·039	291 1178·4 2·051	299 1187·4 2·065	308 1196·6 2·078	328 1219·6 2·106	348 1242·3 2·134
26·5	1·714 (120·5)	201 1114·1 1·933	— — —	— — —	204 1119·2 1·942	208 1123·7 1·950	211 1128·2 1·957	215 1132·7 1·964	218 1137·2 1·971	222 1141·8 1·978	225 1146·2 1·986	229 1150·8 1·993	236 1160·0 2·007	243 1169·0 2·020	250 1178·3 2·033	257 1187·4 2·047	263 1196·6 2·060	281 1219·6 2·089	298 1242·3 2·116
26	1·958 (125·3)	177 1116·0 1·922	— — —	— — —	179 1119·0 1·926	182 1123·6 1·934	185 1128·0 1·941	188 1132·6 1·949	191 1137·0 1·956	194 1141·7 1·963	197 1146·1 1·970	200 1150·7 1·977	206 1159·9 1·991	213 1168·9 2·005	219 1178·2 2·018	225 1187·3 2·030	231 1196·5 2·044	246 1219·5 2·072	261 1242·2 2·100
		Sat.	**140°F.**	**150°F.**	**160°F.**	**170°F.**	**180°F.**	**190°F.**	**200°F.**	**210°F.**	**220°F.**	**230°F.**	**240°F.**	**250°F.**	**300°F.**	**350°F.**	**400°F.**	**450°F.**	**500°F.**
25	2·448 (133·6)	144 1119·6 1·904	145 1123·1 1·908	148 1127·7 1·916	150 1132·2 1·923	153 1136·8 1·931	155 1141·4 1·938	158 1145·9 1·945	160 1150·5 1·952	162 1155·0 1·959	165 1159·7 1·966	167 1164·3 1·973	170 1168·8 1·979	172 1173·5 1·986	184 1196·4 2·019	197 1219·4 2·048	209 1242·2 2·075	221 1265·2 2·101	233 1288·7 2·126
24	2·938 (140·7)	121 1122·6 1·889	— — —	123 1127·4 1·896	125 1132·0 1·903	127 1136·4 1·911	129 1141·1 1·918	131 1145·6 1·925	133 1150·2 1·932	135 1154·7 1·939	137 1159·5 1·946	139 1164·1 1·953	141 1168·6 1·959	143 1173·3 1·966	154 1196·3 1·998	164 1219·4 2·028	174 1242·1 2·055	184 1265·2 2·082	194 1288·6 2·106
23	3·428 (146·8)	105 1125·0 1·876	— — —	105 1127·0 1·878	107 1131·6 1·885	109 1136·1 1·893	111 1140·8 1·900	112 1145·3 1·907	114 1150·0 1·914	116 1154·5 1·921	118 1159·2 1·928	119 1163·9 1·935	121 1168·4 1·942	123 1173·2 1·949	132 1196·2 1·981	140 1219·2 2·011	149 1242·0 2·038	158 1265·1 2·064	167 1288·6 2·090
22	3·917 (152·1)	92·6 1127·2 1·865	— — —	— — —	93·6 1131·3 1·870	95·2 1135·9 1·878	96·7 1140·5 1·885	98·2 1145·1 1·892	99·7 1149·8 1·900	101 1154·3 1·907	103 1159·0 1·914	105 1163·7 1·921	106 1168·3 1·928	108 1173·0 1·935	115 1196·2 1·966	123 1219·1 1·996	131 1242·0 2·023	138 1265·0 2·050	146 1288·5 2·075
21	4·407 (157·0)	82·9 1129·1 1·855	— — —	— — —	83·2 1131·0 1·857	84·6 1135·6 1·864	86·0 1140·2 1·872	87·3 1144·8 1·879	88·7 1149·5 1·886	90·1 1154·0 1·893	91·5 1158·8 1·900	92·9 1163·5 1·907	94·2 1168·1 1·915	95·6 1172·9 1·922	102 1196·1 1·953	109 1219·0 1·982	116 1241·9 2·010	123 1265·0 2·036	130 1288·5 2·062

TABLE II. THE PROPERTIES OF SUPERHEATED STEAM—*continued*

VOLUME : CU. FT. PER LB.
TOTAL HEAT : BTU PER LB.
ENTROPY : BTU PER °F. ABS. PER LB.
} AT TEMPERATURES OF :—

PRESSURE: VACUUM IN. HG.	PRESSURE: PSI ABS. (SAT. TEMP.)	Sat.	170°F.	180°F.	190°F.	200°F.	210°F.	220°F.	230°F.	240°F.	260°F.	280°F.	300°F.	320°F.	340°F.	360°F.	380°F.	400°F.	500°F.
20	4·896 (161·4)	75·1 1131·0 1·847	76·1 1135·3 1·853	77·3 1139·9 1·860	78·6 1144·6 1·867	79·8 1149·2 1·874	81·0 1153·8 1·881	82·2 1158·6 1·889	83·5 1163·3 1·896	84·7 1168·0 1·903	87·2 1177·4 1·916	89·6 1186·6 1·929	92·1 1196·0 1·941	94·5 1205·2 1·954	97·0 1214·3 1·966	99·4 1223·6 1·977	102 1232·7 1·988	104 1241·8 1·999	117 1288·4 2·050
19	5·386 (165·4)	68·7 1132·7 1·839	69·2 1135·0 1·842	70·3 1139·6 1·849	71·4 1144·3 1·856	72·5 1149·0 1·864	73·6 1153·6 1·871	74·8 1158·4 1·878	75·9 1163·1 1·885	77·0 1167·8 1·892	79·3 1177·2 1·905	81·5 1186·5 1·918	83·7 1195·9 1·930	85·9 1205·1 1·943	88·2 1214·3 1·955	90·4 1223·5 1·966	92·6 1232·6 1·977	94·8 1241·7 1·988	106 1288·4 2·039
18	5·876 (169·2)	63·3 1134·2 1·832	— — —	64·4 1139·3 1·839	65·5 1144·0 1·847	66·5 1148·8 1·854	67·5 1153·4 1·861	68·5 1158·1 1·868	69·6 1162·9 1·875	70·6 1167·6 1·882	72·7 1177·0 1·895	74·7 1186·3 1·908	76·7 1195·8 1·921	78·8 1205·0 1·934	80·8 1214·2 1·945	82·8 1223·4 1·956	84·9 1232·5 1·967	86·9 1241·7 1·978	97·1 1288·3 2·029
17	6·365 (172·7)	58·8 1135·6 1·825	— — —	59·2 1139·0 1·830	60·1 1143·8 1·838	61·1 1148·5 1·845	62·0 1153·2 1·852	63·0 1157·9 1·859	63·9 1162·7 1·866	64·9 1167·4 1·873	66·8 1176·8 1·886	68·7 1186·2 1·899	70·6 1195·6 1·912	72·4 1204·9 1·924	74·3 1214·1 1·936	76·2 1223·3 1·947	78·1 1232·4 1·958	80·0 1241·6 1·968	89·4 1288·2 2·020
16	6·855 (175·9)	54·8 1136·9 1·819	— — —	55·5 1138·7 1·822	56·4 1143·5 1·829	57·3 1148·0 1·837	58·2 1153·0 1·844	59·1 1157·7 1·851	60·0 1162·5 1·858	60 9 1167·3 1·865	62·7 1176·7 1·878	64·5 1186·1 1·891	66·2 1195·5 1·904	68·0 1204·8 1·916	69·8 1214·0 1·928	71·6 1223·3 1·939	73·3 1232·4 1·950	75·1 1241·5 1·960	83·9 1288·2 2·012
		Sat.	190°F.	200°F.	210°F.	220°F.	230°F.	240°F.	250°F.	260°F.	280°F.	300°F.	320°F.	340°F.	360°F.	380°F.	400°F.	450°F.	500°F.
15	7·344 (178·9)	51·4 1138·1 1·813	52·2 1143·2 1·821	53·0 1148·0 1·829	53·8 1152·8 1·836	54·7 1157·5 1·843	55·5 1162·3 1·850	56·4 1167·1 1·857	57·2 1171·9 1·864	58·0 1176·6 1·871	59·7 1186·0 1·883	61·3 1195·4 1·896	63·0 1204·7 1·908	64·6 1213·9 1·920	66·2 1223·2 1·932	67·9 1232·3 1·943	69·5 1241·5 1·953	73·6 1264·7 1·980	77·7 1288·1 2·005
14	7·834 (181·9)	48·4 1139·2 1·808	48·9 1143·0 1·814	49·7 1147·8 1·821	50·4 1152·6 1·829	51·2 1157·3 1·836	52·0 1162·1 1·843	52·8 1167·0 1·850	53·6 1171·7 1·857	54·4 1176·4 1·863	55·9 1185·8 1·876	57·4 1195·3 1·889	59·0 1204·6 1·901	60·5 1213·8 1·913	62·1 1223·1 1·925	63·6 1232·2 1·936	65·2 1241·4 1·946	69·0 1264·6 1·972	72·8 1288·1 1·997
13	8·324 (184·7)	45·7 1140·4 1·803	46·0 1142·7 1·807	46·8 1147·5 1·815	47·5 1152·4 1·822	48·2 1157·1 1·829	48·9 1162·0 1·836	49·7 1166·8 1·843	50·4 1171·6 1·850	51·2 1176·2 1·857	52·6 1185·7 1·870	54·1 1195·2 1·882	55·5 1204·5 1·894	57·0 1213·8 1·906	58·4 1223·0 1·918	59·9 1232·1 1·929	61·3 1241·3 1·939	64·9 1264·6 1·965	68·5 1288·0 1·990
12	8·813 (187·3)	43·3 1141·4 1·798	43·4 1142·4 1·801	44·1 1147·2 1·808	44·8 1152·2 1·815	45·5 1156·9 1·822	46·2 1161·8 1·830	46·9 1166·6 1·837	47·6 1171·4 1·844	48·3 1176·1 1·850	49·6 1185·6 1·863	51·0 1195·1 1·876	52·4 1204·4 1·888	53·8 1213·6 1·900	55·1 1222·9 1·912	56·5 1232·0 1·923	57·9 1241·2 1·933	61·3 1264·5 1·959	64·6 1288·0 1·984
11	9·303 (189·8)	41·2 1142·5 1·794	— — —	41·8 1147·0 1·802	42·4 1152·0 1·809	43·1 1156·7 1·816	43·8 1161·6 1·823	44·4 1166·4 1·831	45·1 1171·2 1·838	45·7 1176·0 1·845	47·0 1185·5 1·858	48·3 1195·0 1·870	49·6 1204·3 1·882	50·9 1213·6 1·894	52·2 1222·8 1·906	53·5 1231·9 1·917	54·8 1241·2 1·927	58·1 1264·4 1·954	61·2 1287·9 1·978

TABLE II. THE PROPERTIES OF SUPERHEATED STEAM—*continued*

VOLUME: CU. FT. PER LB.; TOTAL HEAT: BTU PER LB.; ENTROPY: BTU PER °F. ABS. PER LB.; AT TEMPERATURES OF:—

PRESSURE VACUUM IN. HG.	PSI ABS. (SAT. TEMP.)	Sat.	200°F.	210°F.	220°F.	230°F.	240°F.	250°F.	260°F.	280°F.	300°F.	320°F.	340°F.	360°F.	380°F.	400°F.	450°F.	500°F.	600°F.
10	9·793 (192·2)	39·2 1143·4 1·790	39·6 1146·8 1·796	40·3 1151·8 1·804	40·9 1156·5 1·811	41·5 1161·5 1·818	42·2 1166·3 1·825	42·8 1171·1 1·832	43·4 1175·8 1·839	44·7 1185·3 1·852	45·9 1194·8 1·864	47·1 1204·2 1·876	48·4 1213·5 1·888	49·6 1222·7 1·900	50·8 1231·8 1·911	52·1 1241·1 1·921	55·1 1264·4 1·948	58·2 1287·9 1·973	64·8 1335·3 2·020
9	10·28 (194·5)	37·5 1144·3 1·786	37·8 1146·6 1·791	38·4 1151·6 1·798	39·0 1156·3 1·805	39·6 1161·3 1·812	40·2 1166·1 1·819	40·8 1170·9 1·826	41·3 1175·6 1·833	42·5 1185·2 1·846	43·7 1194·7 1·858	44·9 1204·1 1·870	46·1 1213·3 1·882	47·3 1222·6 1·894	48·4 1231·7 1·905	49·6 1241·0 1·916	52·5 1264·3 1·942	55·4 1287·8 1·967	61·2 1335·8 2·015
8	10·77 (196·8)	35·9 1145·2 1·782	36·0 1146·4 1·786	36·6 1151·4 1·793	37·2 1156·1 1·800	37·7 1161·1 1·807	38·3 1166·0 1·814	38·9 1170·8 1·821	39·4 1175·5 1·828	40·6 1185·0 1·841	41·7 1194·6 1·853	42·8 1203·9 1·865	44·0 1213·2 1·877	45·1 1222·5 1·889	46·2 1231·6 1·900	47·3 1240·9 1·911	50·2 1264·2 1·936	52·9 1287·8 1·962	58·4 1335·8 2·010
7	11·26 (198·9)	34·4 1146·1 1·779	34·5 1146·2 1·780	35·0 1151·2 1·788	35·6 1156·9 1·795	36·1 1160·9 1·802	36·6 1165·8 1·809	37·2 1170·6 1·816	37·7 1175·4 1·822	38·8 1184·9 1·835	39·9 1194·4 1·848	41·0 1203·8 1·860	42·1 1213·1 1·872	43·1 1222·4 1·884	44·2 1231·6 1·895	45·3 1240·9 1·905	48·0 1264·2 1·932	50·6 1287·7 1·957	55·9 1335·7 2·005
6	11·75 (201·0)	33·1 1146·8 1·775	— — —	33·5 1151·0 1·782	34·0 1155·8 1·790	34·6 1160·8 1·797	35·1 1165·6 1·804	35·6 1170·4 1·811	36·1 1175·2 1·818	37·2 1184·7 1·830	38·2 1194·3 1·843	39·3 1203·7 1·855	40·3 1213·0 1·867	41·3 1222·3 1·879	42·4 1231·5 1·890	43·4 1240·8 1·901	46·0 1264·1 1·927	48·5 1287·7 1·952	53·6 1335·7 2·000
		Sat.	**210°F.**	**220°F.**	**230°F.**	**240°F.**	**250°F.**	**260°F.**	**270°F.**	**280°F.**	**290°F.**	**300°F.**	**320°F.**	**340°F.**	**360°F.**	**380°F.**	**400°F.**	**500°F.**	**600°F.**
5	12·24 (203·0)	31·9 1147·5 1·772	32·2 1150·8 1·778	32·7 1155·6 1·785	33·2 1160·6 1·792	33·7 1165·5 1·799	34·2 1170·3 1·806	34·7 1175·0 1·813	35·2 1179·9 1·819	35·7 1184·6 1·826	36·2 1189·4 1·832	36·7 1194·2 1·839	37·7 1203·6 1·851	38·7 1212·9 1·863	39·7 1222·2 1·874	40·6 1231·4 1·885	41·6 1240·7 1·896	46·5 1287·6 1·948	51·4 1335·7 1·996
4	12·73 (204·9)	30·7 1148·2 1·769	30·9 1150·7 1·774	31·4 1155·5 1·781	31·9 1160·4 1·788	32·4 1165·3 1·795	32·8 1170·1 1·802	33·3 1174·9 1·808	33·8 1179·8 1·814	34·3 1184·4 1·821	34·8 1189·2 1·828	35·3 1194·0 1·835	36·2 1203·5 1·846	37·2 1212·8 1·858	38·1 1222·1 1·870	39·1 1231·3 1·881	40·0 1240·7 1·892	44·7 1287·6 1·944	49·4 1335·6 1·991
3	13·22 (206·8)	29·7 1148·8 1·766	29·8 1150·5 1·770	30·2 1155·3 1·776	30·7 1160·3 1·783	31·1 1165·2 1·790	31·6 1170·0 1·797	32·1 1174·7 1·804	32·6 1179·6 1·811	33·0 1184·2 1·817	33·5 1189·1 1·824	33·9 1193·9 1·830	34·9 1203·4 1·842	35·8 1212·7 1·854	36·7 1222·0 1·865	37·6 1231·2 1·876	38·5 1240·6 1·887	43·1 1287·5 1·940	47·6 1335·6 1·987
2	13·71 (208·6)	28·7 1149·5 1·763	28·7 1150·3 1·765	29·1 1155·2 1·772	29·6 1160·1 1·779	30·0 1165·0 1·786	30·5 1169·8 1·793	30·9 1174·6 1·799	31·4 1179·5 1·806	31·8 1184·1 1·812	32·3 1189·0 1·819	32·7 1193·8 1·826	33·6 1203·2 1·838	34·5 1212·6 1·850	35·4 1221·9 1·861	36·3 1231·1 1·872	37·2 1240·5 1·883	41·5 1287·5 1·935	45·9 1335·5 1·983
1	14·20 (210·3)	27·7 1150·1 1·760	— — —	28·1 1155·0 1·768	28·5 1160·0 1·775	29·0 1164·8 1·782	29·4 1169·7 1·789	29·8 1174·4 1·796	30·3 1179·4 1·802	30·7 1184·0 1·808	31·2 1188·8 1·815	31·6 1193·6 1·821	32·4 1203·1 1·833	33·3 1212·5 1·845	34·2 1221·8 1·857	35·0 1231·0 1·868	35·9 1240·4 1·879	40·1 1287·4 1·931	44·3 1335·5 1·979

TABLE II. THE PROPERTIES OF SUPERHEATED STEAM—*continued*

VOLUME : CU. FT. PER LB.
TOTAL HEAT : BTU PER LB.
ENTROPY : BTU PER °F. ABS. PER LB.
} AT TEMPERATURES OF :—

PSI GAUGE	PSI ABS. (SAT. TEMP.)	Sat.	220°F.	230°F.	240°F.	250°F.	260°F.	270°F.	280°F.	290°F.	300°F.	320°F.	340°F.	360°F.	380°F.	400°F.	450°F.	500°F.	600°F.
0	14·696 (212)	26·8 1150·8 1·757	27·1 1154·9 1·764	27·6 1159·8 1·771	28·0 1164·6 1·778	28·4 1169·5 1·785	28·8 1174·3 1·791	29·2 1179·2 1·798	29·7 1183·8 1·805	30·1 1188·7 1·811	30·5 1193·5 1·818	31·3 1202·9 1·830	32·2 1212·4 1·841	33·0 1221·7 1·853	33·8 1230·9 1·865	34·7 1240·3 1·875	36·7 1263·7 1·902	38·8 1287·4 1·927	42·9 1335·4 1·975
1	15·7 (215·4)	25·2 1152·0 1·752	25·4 1154·6 1·757	25·8 1159·5 1·763	26·2 1164·3 1·770	26·6 1169·2 1·777	27·0 1174·0 1·784	27·4 1178·9 1·791	27·7 1183·5 1·797	28·1 1188·3 1·803	28·5 1193·2 1·810	29·3 1202·6 1·822	30·1 1212·1 1·833	30·9 1221·4 1·845	31·6 1230·7 1·857	32·4 1240·2 1·868	34·3 1263·5 1·894	36·3 1287·3 1·920	40·1 1335·3 1·968
2	16·7 (218·5)	23·8 1153·2 1·747	23·8 1154·3 1·749	24·2 1159·2 1·756	24·6 1164·0 1·763	25·0 1168·9 1·770	25·3 1173·6 1·776	25·7 1178·6 1·783	26·1 1183·2 1·790	26·4 1188·0 1·796	26·8 1192·8 1·802	27·5 1202·3 1·814	28·3 1211·8 1·826	29·0 1221·2 1·838	29·7 1230·5 1·850	30·5 1240·0 1·861	32·3 1263·4 1·887	34·1 1287·1 1·912	37·7 1335·2 1·960
3	17·7 (221·5)	22·5 1154·3 1·742	— — —	22·8 1158·9 1·749	23·2 1163·6 1·756	23·5 1168·6 1·763	23·9 1173·3 1·769	24·2 1178·2 1·776	24·6 1182·9 1·783	24·9 1187·8 1·789	25·3 1192·5 1·796	26·0 1202·1 1·808	26·7 1211·6 1·820	27·4 1220·9 1·832	28·1 1230·3 1·843	28·7 1239·8 1·854	30·5 1263·2 1·881	32·2 1286·9 1·906	35·6 1335·1 1·954
4	18·7 (224·5)	21·4 1155·3 1·738	— — —	21·6 1158·6 1·743	21·9 1163·4 1·750	22·3 1168·2 1·757	22·6 1173·0 1·763	22·9 1177·9 1·770	23·3 1182·6 1·776	23·6 1187·4 1·783	23·9 1192·2 1·790	24·6 1201·7 1·802	25·2 1211·3 1·813	25·9 1220·7 1·825	26·5 1230·1 1·837	27·2 1239·6 1·848	28·8 1263·0 1·875	30·4 1286·8 1·900	33·7 1334·9 1·948

PSI GAUGE	PSI ABS. (SAT. TEMP.)	Sat.	230°F.	240°F.	250°F.	260°F.	270°F.	280°F.	290°F.	300°F.	320°F.	340°F.	360°F.	380°F.	400°F.	450°F.	500°F.	550°F.	600°F.
5	19·7 (227·4)	20·4 1156·3 1·734	20·5 1158·4 1·737	20·8 1163·1 1·744	21·1 1167·9 1·751	21·4 1172·7 1·757	21·8 1177·6 1·764	22·1 1182·3 1·770	22·4 1187·1 1·777	22·7 1191·9 1·783	23·3 1201·5 1·796	23·9 1211·0 1·807	24·6 1220·4 1·819	25·2 1229·9 1·831	25·8 1239·4 1·842	27·3 1262·8 1·869	28·9 1286·6 1·894	30·4 1310·6 1·919	31·9 1334·8 1·942
6	20·7 (230·0)	19·4 1157·3 1·730	— — —	19·8 1162·8 1·738	20·1 1167·6 1·745	20·4 1172·3 1·751	20·7 1177·3 1·758	21·0 1182·0 1·764	21·3 1186·8 1·771	21·6 1191·6 1·777	22·2 1201·2 1·790	22·8 1210·8 1·802	23·4 1220·2 1·814	24·0 1229·8 1·825	24·6 1239·2 1·836	26·0 1262·8 1·863	27·5 1286·4 1·888	28·9 1310·6 1·913	30·4 1334·8 1·936
7	21·7 (232·4)	18·6 1158·2 1·726	— — —	18·8 1162·4 1·732	19·1 1167·3 1·739	19·4 1172·1 1·746	19·7 1177·0 1·753	20·0 1181·8 1·759	20·3 1186·6 1·765	20·6 1191·4 1·772	21·1 1201·0 1·785	21·7 1210·6 1·796	22·3 1220·0 1·808	22·8 1229·7 1·820	23·4 1239·1 1·831	24·8 1262·7 1·858	26·2 1286·3 1·883	27·6 1310·5 1·908	29·0 1334·7 1·931
8	22·7 (234·8)	17·9 1159·1 1·722	— — —	18·0 1162·1 1·727	18·3 1167·0 1·734	18·6 1171·8 1·740	18·9 1176·8 1·747	19·1 1181·6 1·754	19·4 1186·3 1·760	19·7 1191·2 1·766	20·2 1200·8 1·779	20·8 1210·4 1·791	21·3 1219·9 1·803	21·9 1229·5 1·815	22·4 1238·9 1·826	23·7 1262·6 1·852	25·1 1286·2 1·878	26·4 1310·4 1·903	27·7 1334·6 1·926
9	23·7 (237·1)	17·2 1160·0 1·719	— — —	17·2 1161·8 1·721	17·5 1166·8 1·728	17·8 1171·6 1·735	18·0 1176·5 1·742	18·3 1181·3 1·749	18·6 1186·1 1·755	18·8 1190·9 1·761	19·3 1200·6 1·774	19·9 1210·2 1·786	20·4 1219·7 1·798	20·9 1229·4 1·810	21·4 1238·8 1·820	22·7 1262·5 1·847	24·0 1286·2 1·873	25·3 1310·4 1·897	26·5 1334·5 1·921

TABLE II. THE PROPERTIES OF SUPERHEATED STEAM—*continued*

VOLUME : CU. FT. PER LB.
TOTAL HEAT : BTU PER LB.
ENTROPY : BTU PER °F. ABS. PER LB.
} AT TEMPERATURES OF :—

PRESSURE PSI GAUGE	PSI ABS. (SAT. TEMP.)	Sat.	250°F.	260°F.	270°F.	280°F.	290°F.	300°F.	310°F.	320°F.	330°F.	340°F.	360°F.	380°F.	400°F.	450°F.	500°F.	550°F.	600°F.
10	24·7 (239·4)	16·5 1160·8 1·716	16·8 1166·5 1·723	17·0 1171·3 1·730	17·3 1176·3 1·738	17·5 1181·1 1·744	17·8 1185·9 1·750	18·1 1190·7 1·757	18·3 1195·6 1·763	18·6 1200·4 1·769	18·8 1205·2 1·775	19·1 1210·0 1·782	19·6 1219·5 1·793	20·1 1229·2 1·805	20·6 1238·6 1·816	21·8 1262·4 1·843	23·0 1286·1 1·869	24·2 1310·2 1·893	25·5 1334·5 1·917
11	25·7 (241·6)	15·9 1161·6 1·712	16·1 1166·2 1·719	16·4 1171·1 1·726	16·6 1176·0 1·733	16·8 1180·8 1·740	17·1 1185·7 1·746	17·3 1190·5 1·753	17·6 1195·3 1·759	17·8 1200·2 1·765	18·1 1205·0 1·771	18·3 1209·8 1·777	18·8 1219·4 1·789	19·3 1229·0 1·801	19·8 1238·5 1·812	20·9 1262·2 1·838	22·1 1286·0 1·864	23·3 1310·1 1·888	24·5 1334·4 1·912
12	26·7 (243·7)	15·3 1162·3 1·709	15·5 1166·0 1·714	15·7 1170·9 1·721	16·0 1175·8 1·728	16·2 1180·6 1·735	16 4 1185·4 1·741	16·7 1190·2 1·748	16·9 1195·1 1·754	17·2 1200 0 1·760	17·4 1204·8 1·767	17·6 1209·6 1·773	18·1 1219·2 1·785	18·5 1228 9 1·796	19·0 1238·4 1·808	20·1 1262·1 1·834	21·3 1286·0 1·860	22·4 1310·0 1·884	23·5 1334·3 1·908
13	27·7 (245·8)	14·8 1163·0 1·706	14·9 1165·7 1·710	15·2 1170·6 1·717	15·4 1175·5 1·724	15·6 1180·4 1·731	15·8 1185·2 1·737	16·1 1190·0 1·743	16·3 1194·9 1·750	16·5 1199·8 1·756	16·8 1204·6 1·762	17·0 1209·4 1·768	17·4 1219·0 1·780	17·9 1228·7 1·792	18·3 1238·2 1·803	19 4 1262·0 1·830	20·5 1285·9 1·856	21·6 1310·0 1·880	22·7 1334·2 1·903
14	28·7 (247·9)	14·3 1163·7 1·703	14·4 1165·4 1·706	14·6 1170·3 1·712	14·9 1175·3 1·719	15·1 1180·1 1·726	15·3 1185·0 1·733	15 5 1189·8 1·739	15·7 1194·7 1·745	15 9 1199·6 1·752	16·2 1204·4 1·758	16·4 1209·2 1·764	16·8 1218·8 1·776	17·2 1228 6 1·787	17·7 1238·1 1·799	18·7 1261·9 1·825	19·8 1285·8 1·852	20·9 1309·9 1·875	21·9 1334·2 1·899
		Sat.	**260°F.**	**270°F.**	**280°F.**	**290°F.**	**300°F.**	**310°F.**	**320°F.**	**330°F.**	**340°F.**	**350°F.**	**360°F.**	**380°F.**	**400°F.**	**450°F.**	**500°F.**	**550°F.**	**600°F.**
15	29·7 (249·8)	13·9 1164·4 1·700	14·1 1170·1 1·708	14·3 1175·0 1·715	14·6 1179·9 1·722	14·8 1184·8 1·729	15·0 1189·6 1·735	15·2 1194·5 1·741	15·4 1199·4 1·748	15·6 1204·2 1·754	15·8 1209·1 1·760	16·0 1213·9 1·766	16·2 1218·6 1·772	16·7 1228·4 1·783	17·1 1237·9 1·795	18·1 1261·8 1·821	19·1 1285·8 1·848	20·1 1309·8 1·871	21·2 1334·1 1·895
16	30·7 (251·7)	13·4 1165·1 1·698	13·7 1169·9 1·704	13·9 1174·8 1·711	14·1 1179·7 1·718	14·3 1184·5 1·725	14·5 1189·3 1·731	14·7 1194·2 1·737	14·9 1199·2 1·744	15·1 1204·0 1·750	15·3 1208·9 1·756	15·5 1213·7 1·762	15·7 1218·5 1·768	16·1 1228·2 1·780	16·5 1237·8 1·791	17·5 1261·7 1·818	18·5 1285·7 1·843	19·5 1309·7 1·868	20·5 1334·0 1·891
17	31·7 (253·6)	13·0 1165·7 1·696	13·2 1169·6 1·700	13·4 1174·6 1·707	13·6 1179·4 1·714	13·8 1184·3 1·721	14·0 1189·1 1·728	14·2 1194·0 1·734	14 4 1199·0 1·740	14·6 1203·8 1·746	14·8 1208·7 1·752	15·0 1213·5 1·758	15·2 1218·3 1·764	15·6 1228·1 1·776	16·0 1237·7 1·787	16·9 1261·6 1·814	17·9 1285·6 1·840	18·9 1309·6 1·864	19·8 1334·0 1·888
18	32·7 (255·4)	12·7 1166·4 1·693	12·8 1169·4 1·697	13·0 1174·3 1·704	13·2 1179·2 1·710	13·4 1184·0 1·717	13·6 1188·9 1·724	13·8 1193·8 1·730	13·9 1198·8 1·736	14·1 1203·6 1·742	14·3 1208·5 1·749	14·5 1213·3 1·755	14·7 1218·1 1·761	15·1 1227·9 1·772	15·5 1237·5 1·784	16·4 1261·4 1·810	17·4 1285·5 1·836	18·3 1309·5 1·861	19·2 1333·9 1·884
19	33·7 (257·2)	12·3 1167·0 1·691	12·4 1169·1 1·693	12·6 1174·0 1·700	12·8 1179·0 1·707	13·0 1183·8 1·714	13·2 1188·6 1·720	13·3 1193·6 1·726	13·5 1198·6 1·733	13·7 1203·4 1·739	13·9 1208·3 1·745	14·1 1213·1 1·751	14·3 1218·0 1·757	14·6 1227·8 1·769	15·0 1237·4 1·780	15·9 1261·3 1·807	16·8 1285·4 1·833	17·7 1309·4 1·857	18·6 1333·8 1·881

TABLE II. THE PROPERTIES OF SUPERHEATED STEAM—*continued*

VOLUME : CU. FT. PER LB.
TOTAL HEAT : BTU PER LB.
ENTROPY : BTU PER °F. ABS. PER LB.
} AT TEMPERATURES OF :—

PRESSURE PSI GAUGE	PSI ABS. (SAT. TEMP.)	Sat.	270°F.	280°F.	290°F.	300°F.	310°F.	320°F.	330°F.	340°F.	350°F.	360°F.	370°F.	380°F.	400°F.	450°F.	500°F.	550°F.	600°F
20	34·7 (258·8)	12·0 1167·6 1·688	12·2 1173·8 1·697	12·4 1178·7 1·703	12·6 1183·6 1·710	12·8 1188·4 1·717	13·0 1193·4 1·723	13·1 1198·3 1·729	13·3 1203·2 1·735	13·5 1208·1 1·741	13·7 1213·0 1·747	13·9 1217·8 1·753	14·0 1222·7 1·759	14·2 1227·6 1·765	14·6 1237·2 1·777	15·5 1261·2 1·803	16·3 1285·3 1·829	17·2 1309·3 1·854	18·1 1333·8 1·878
22	36·7 (262·3)	11·4 1168·7 1·684	11·5 1173·3 1·690	11·7 1178·2 1·697	11·9 1183·2 1·703	12·1 1188·0 1·710	12·2 1192·9 1·717	12·4 1197·9 1·722	12·6 1202·8 1·729	12·8 1207·8 1·735	12·9 1212·6 1·741	13·1 1217·5 1·747	13·3 1222·4 1·753	13·4 1227·3 1·759	13·8 1236·9 1·770	14·6 1261·0 1·797	15·4 1285·1 1·823	16·3 1309·2 1·848	17·1 1333·6 1·871
24	38·7 (265·3)	10·8 1169·8 1·680	10·9 1172·8 1·683	11·1 1177·8 1·690	11·3 1182·7 1·697	11·4 1187·5 1·703	11·6 1192·5 1·710	11·8 1197·5 1·716	11·9 1202·4 1·722	12·1 1207·4 1·729	12·2 1212·2 1·735	12·4 1217·1 1·740	12·6 1222·0 1·747	12·7 1226·9 1·753	13·1 1236·6 1·764	13·9 1260·8 1·791	14·7 1285·0 1·817	15·4 1309·0 1·841	16·2 1333·5 1·865
26	40·7 (268·3)	10·3 1170·8 1·676	10·4 1172·3 1·677	10·5 1177·2 1·684	10·7 1182·2 1·691	10·9 1187·0 1·697	11·0 1192·1 1·704	11·2 1197·1 1·710	11·3 1202·0 1·716	11·5 1207·0 1·723	11·6 1211·9 1·729	11·8 1216·8 1·735	11·9 1221·7 1·740	12·1 1226·6 1·747	12·4 1236·3 1·758	13·2 1260·5 1·785	13·9 1284·8 1·811	14·7 1308·8 1·836	15·4 1333·3 1·859
28	42·7 (271·4)	9·87 1171·8 1·672	— — —	10·0 1176·8 1·678	10·2 1181·7 1·685	10·3 1186·6 1·692	10·5 1191·6 1·698	10·6 1196·7 1·704	10·8 1201·6 1·711	10·9 1206·6 1·717	11·1 1211·5 1·723	11·2 1216·4 1·729	11·4 1221·4 1·735	11·5 1226·3 1·741	11·8 1236·0 1·753	12·5 1260·3 1·780	13·3 1284·6 1·806	14·0 1308·6 1·830	14·7 1333·2 1·854

PRESSURE PSI GAUGE	PSI ABS. (SAT. TEMP.)	Sat.	280°F.	290°F.	300°F.	310°F.	320°F.	330°F.	340°F.	350°F.	360°F.	370°F.	380°F.	390°F.	400°F.	450°F.	500°F.	550°F.	600°F.
30	44·7 (274·0)	9·46 1172·7 1·668	9·56 1176·3 1·673	9·71 1181·2 1·679	9·85 1186·2 1·686	10·0 1191·2 1·693	10·1 1196·2 1·699	10·3 1201·2 1·705	10·4 1206·2 1·712	10·6 1211·2 1·718	10·7 1216·1 1·724	10·9 1221·1 1·730	11·0 1226·0 1·736	11·1 1230·8 1·742	11·3 1235·8 1·747	12·0 1260·1 1·775	12·7 1284·3 1·800	13·3 1308·5 1·825	14·0 1333·0 1·849
32	46·7 (276·7)	9·08 1173·5 1·665	9·14 1175·8 1·667	9·28 1180·8 1·674	9·42 1185·7 1·681	9·56 1190·8 1·687	9·70 1195·9 1·694	9·83 1200·8 1·700	9·98 1205·9 1·706	10·1 1210·8 1·713	10·2 1215·8 1·719	10·4 1220·7 1·725	10·5 1225·6 1·731	10·7 1230·5 1·737	10·8 1235·4 1·742	11·5 1259·8 1·770	12·1 1284·1 1·796	12·8 1308·3 1·820	13·4 1332·8 1·844
34	48·7 (279·4)	8·73 1174·3 1·661	— — —	8·89 1180·3 1·669	9·02 1185·3 1·676	9·16 1190·3 1·682	9·29 1195·4 1·689	9·43 1200·4 1·695	9·56 1205·5 1·701	9·69 1210·4 1·708	9·82 1215·4 1·714	9·95 1220·4 1·720	10·1 1225·3 1·726	10·2 1230·2 1·732	10·3 1235·2 1·737	11·0 1259·6 1·765	11·6 1283·9 1·791	12·2 1308·1 1·815	12·9 1332·6 1·839
36	50·7 (281·9)	8·40 1175·1 1·658	— — —	8·52 1179·8 1·664	8·65 1184·8 1·670	8·78 1189·9 1·677	8·91 1195·0 1·684	9·04 1200·1 1·690	9·17 1205·1 1·697	9·29 1210·1 1·703	9·42 1215·1 1·709	9·55 1220·1 1·715	9·67 1225·0 1·721	9·80 1229·9 1·727	9·91 1234·9 1·732	10·5 1259·4 1·760	11·1 1283·7 1·786	11·8 1307·9 1·811	12·3 1332·5 1·834
38	52·7 (284·4)	8·11 1175·8 1·655	— — —	8·19 1179·3 1·659	8·32 1184·4 1·666	8·44 1189·5 1·673	8·56 1194·6 1·679	8·69 1199·7 1·685	8·81 1204·7 1·692	8·93 1209·8 1·698	9·06 1214·8 1·704	9·18 1219·8 1·710	9·30 1224·7 1·716	9·42 1229·6 1·722	9·54 1234·6 1·728	10·1 1259·1 1·756	10·7 1283·5 1·782	11·3 1307·7 1·806	11·9* 1332·3 1·830

TABLE II. THE PROPERTIES OF SUPERHEATED STEAM—*continued*

VOLUME : CU. FT. PER LB.; TOTAL HEAT : BTU PER LB.; ENTROPY : BTU PER °F. ABS. PER LB. } AT TEMPERATURES OF :—

PRESSURE PSI GAUGE	PSI ABS. (SAT. TEMP.)	Sat.	300°F.	310°F.	320°F.	330°F.	340°F.	350°F.	360°F.	370°F.	380°F.	390°F.	400°F.	420°F.	440°F.	460°F.	500°F.	550°F.	600°F.
40	54·7 (286·7)	7·83 1176·5 1·652	8·03 1184·0 1·661	8·15 1189·1 1·668	8·27 1194·2 1·675	8·39 1199·3 1·681	8·51 1204·3 1·687	8·63 1209·4 1·694	8·75 1214·4 1·700	8·87 1219·4 1·706	8·98 1224·4 1·712	9·10 1229·3 1·718	9·22 1234·3 1·724	9·45 1244·2 1·735	9·69 1254·0 1·746	9·91 1263·8 1·757	10·4 1283·3 1·777	10·9 1307·6 1·802	11·5 1332·1 1·826
42	56·7 (289·0)	7·57 1177·1 1·649	7·73 1183 6 1·657	7·85 1188·7 1·663	7·97 1193·8 1·670	8·08 1198·9 1·676	8·20 1204 0 1·683	8·31 1209·0 1·689	8·43 1214·1 1·696	8·54 1219·1 1·701	8·65 1224·0 1·707	8·77 1229·0 1·713	8·88 1234·0 1·719	9·10 1243·9 1·730	9·33 1253·7 1·741	9·55 1263·6 1·752	9·99 1283·1 1·773	10·5 1307·4 1·798	11·1 1332·0 1·821
44	58·7 (291·3)	7·33 1177·8 1·646	7·46 1183·2 1·652	7·57 1188·3 1·659	7·68 1193·4 1·666	7·80 1198·5 1·672	7·91 1203·6 1·679	8·02 1208·7 1·685	8·13 1213·8 1·691	8·24 1218·7 1·697	8·35 1223·8 1·703	8·46 1228·7 1·709	8·57 1233·7 1·715	8·78 1243·6 1·726	9·00 1253·5 1·737	9·21 1263·3 1·748	9·64 1282·9 1·769	10·2 1307·2 1·794	10·7 1331·8 1·818
46	60·7 (293·5)	7·10 ·178·4 1·644	7 20 1182·6 1·648	7·31 1187·8 1·655	7·42 1192·9 1·662	7·53 1198·2 1·668	7·64 1203·2 1·675	7·75 1208·3 1·681	7·85 1213·4 1·687	7·96 1218·4 1·693	8·07 1223·5 1·699	8·17 1228·4 1·705	8·28 1233·4 1·711	8·51 1243·4 1·722	8·70 1253·2 1·733	8·90 1263·1 1·744	9·31 1282·7 1·765	9·83 1307·0 1·790	10·3 1331·7 1·814
48	62·7 (295·6)	6·89 1179·0 1·641	6·96 1182·1 1·644	7·07 1187·3 1·651	7·18 1192·5 1·658	7·29 1197·8 1·664	7·39 1202·9 1·671	7·50 1208·0 1·677	7·60 1213·0 1·683	7·70 1218·1 1·689	7·80 1223·1 1·695	7·91 1228·1 1·701	8·01 1233·1 1·707	8·22 1243·1 1·718	8·42 1252·9 1·730	8·62 1262·9 1·740	9·02 1282·5 1·762	9·51 1306·8 1·786	10·0 1331·6 1·810
		Sat.	**310°F.**	**320°F.**	**330°F.**	**340°F.**	**350°F.**	**360°F.**	**370°F.**	**380°F.**	**390°F.**	**400°F.**	**420°F.**	**440°F.**	**460°F.**	**480°F.**	**500°F.**	**550°F.**	**600°F.**
50	64·7 (297·7)	6·68 1179·6 1 638	6·82 1186·9 1·647	6·93 1192·1 1·654	7·03 1197·4 1·661	7 13 1202 5 1·667	7·23 1207·6 1·673	7·33 1212·6 1·680	7·44 1217·8 1·686	7·54 1222·8 1·692	7·63 1227·8 1·698	7·73 1232·8 1·704	7·93 1242·8 1·715	8·13 1252·7 1·726	8·32 1262·6 1·737	8·52 1272·4 1·747	8·71 1282·3 1·758	9·18 1306·6 1·783	9·66 1331·4 1·806
55	69·7 (302·7)	6·24 1181·0 1·633	6·31 1185·8 1·638	6·41 1191·1 1·645	6·50 1196·4 1·652	6 60 1201·5 1·658	6·69 1206·7 1 665	6·79 1211·8 1 671	6·88 1216·9 1·677	6·98 1222·0 1·683	7·07 1227·0 1·689	7·16 1232·0 1·695	7·35 1242·1 1·707	7·53 1252·0 1·718	7·71 1262·1 1·729	7·89 1271·9 1·739	8·07 1281·8 1·749	8·52 1306·2 1·775	8·95 1331·1 1·799
60	74·7 (307·4)	5·84 1182·4 1·627	5·87 1184·7 1·630	5·96 1190·1 1·637	6·05 1195·3 1·643	6·15 1200·5 1·650	6·24 1205·8 1·656	6·32 1210·9 1·663	6·41 1216·1 1·669	6·50 1221·2 1·675	6·59 1226·2 1·681	6·67 1231·3 1·687	6·84 1241·4 1·699	7·01 1251·4 1·710	7·18 1261·5 1·721	7·36 1271·4 1·731	7·52 1281·4 1·741	7·94 1305·9 1·767	8·35 1330·7 1·791
65	79·7 (311·8)	5·50 1183·7 1·622	— — —	5·57 1189·0 1·628	5·66 1194·2 1·635	5·74 1199·5 1·642	5·82 1204·8 1·649	5·91 1210·0 1·655	5·99 1215·2 1·661	6·08 1220·4 1·667	6·16 1225·4 1·673	6·24 1230·5 1·679	6·40 1240·8 1·691	6·56 1250·8 1·702	6·72 1260·9 1·714	6·88 1270·8 1·724	7·04 1280·9 1·734	7·43 1305·5 1·760	7·82 1330·4 1·783
70	84·7 (316·0)	5·19 1185·0 1·617	— — —	5·23 1187·9 1·620	5·31 1193·1 1·627	5·39 1198·5 1·634	5·47 1203·8 1·641	5·55 1209·0 1·647	5·63 1214·3 1·654	5·71 1219·5 1·660	5·79 1224·6 1·666	5·87 1229·8 1·672	6·02 1240·0 1·684	6·17 1250·1 1·695	6·32 1260·3 1·706	6·47 1270·3 1·717	6·62 1280·4 1·727	6·99 1305·1 1·753	7·36 1330·0 1·777

TABLE II. THE PROPERTIES OF SUPERHEATED STEAM—*continued*

PRESSURE		VOLUME : CU. FT. PER LB. TOTAL HEAT : BTU PER LB. ENTROPY : BTU PER °F. ABS. PER LB.	AT TEMPERATURES OF :—																
PSI GAUGE	PSI ABS. (SAT. TEMP.)	Sat.	330°F.	340°F.	350°F.	360°F.	370°F.	380°F.	390°F.	400°F.	410°F.	420°F.	430°F.	440°F.	460°F.	480°F.	500°F.	550°F.	600°F.
75	89·7 (320·1)	4·92 1186·1 1·612	5·00 1192·0 1·620	5·08 1197·4 1·627	5·16 1202·8 1·634	5·23 1208·1 1·640	5·30 1213·4 1·647	5·38 1218·6 1·653	5·45 1223·8 1·659	5·53 1229·0 1·665	5·60 1234·1 1·671	5·67 1239·4 1·677	5·74 1244·4 1·683	5·81 1249·5 1·688	5·96 1259·8 1·700	6·10 1269·8 1·710	6·24 1279·9 1·720	6·59 1304·8 1·746	6·94 1329·7 1·770
80	94·7 (323·9)	4·67 1187·2 1·608	4·73 1190·8 1·612	4·80 1196·4 1·619	4·87 1201·8 1·626	4·94 1207·1 1·633	5·01 1212·4 1·640	5·08 1217·7 1·646	5·15 1223·0 1·652	5·22 1228·2 1·658	5·29 1233·4 1·664	5·36 1238·6 1·670	5·43 1243·7 1·676	5·50 1248·9 1·681	5·64 1259·2 1·693	5·77 1269·3 1·703	5·91 1279·4 1·714	6·24 1304·4 1·740	6·57 1329·4 1·764
85	99·7 (327·7)	4·45 1188·1 1·604	4·48 1189·6 1·605	4·54 1195·3 1·613	4·61 1200·7 1·620	4·68 1206·1 1·626	4·75 1211·5 1 633	4·82 1216·8 1·640	4·89 1222·2 1·645	4·95 1227·4 1·652	5·02 1232·6 1·658	5·08 1237·9 1·664	5·15 1243·1 1·670	5·22 1248·2 1·675	5·35 1258·6 1·687	5·48 1268·8 1·697	5·61 1279·0 1·708	5·92 1304·0 1·734	6·24 1329·0 1·758
90	104·7 (331·2)	4·25 1189·1 1·600	— — —	4·32 1194·2 1·606	4·38 1199·7 1·613	4·45 1205·2 1·620	4·51 1210·6 1·627	4·58 1216·0 1·633	4·64 1221·4 1·639	4·71 1226·6 1·646	4·77 1231·9 1·652	4·83 1237·2 1·658	4·90 1242·4 1·664	4·96 1247·6 1·669	5·08 1258·0 1·681	5·21 1268·3 1·692	5·33 1278·5 1·702	5·63 1303·7 1·728	5·93 1328·7 1·752
95	109·7 (334·6)	4·07 1189·8 1·596	— — —	4·11 1193·1 1·600	4·17 1198·6 1·607	4·24 1204·2 1·614	4·30 1209·7 1·620	4·36 1215·1 1·627	4·42 1220·5 1·633	4·48 1225·8 1·640	4·54 1231·2 1·646	4·61 1236·5 1·652	4·67 1241·7 1·658	4·73 1247·0 1·664	4·85 1257·4 1·675	4·96 1267·8 1·686	5·08 1278·0 1·697	5·37 1303·0 1·722	5·66 1328·3 1·747
		Sat.	360°F.	370°F.	380°F.	390°F.	400°F.	410°F.	420°F.	430°F.	440°F.	450°F.	460°F.	470°F.	480°F.	500°F.	550°F.	600°F.	700°F.
100	114·7 (337·9)	3·90 1190·6 1·592	4·04 1203·2 1·608	4·10 1208·8 1·615	4·16 1214·1 1·621	4·22 1219·7 1·628	4·28 1225·0 1·634	4·34 1230 4 1·640	4·40 1235·8 1·646	4·46 1241·0 1·652	4·51 1246·3 1·658	4·57 1251·6 1·664	4·63 1256·8 1·670	4·68 1262·0 1·675	4·74 1267·3 1·680	4·85 1277·5 1·691	5·13 1302·9 1·717	5·41 1327·9 1·741	5·95 1377·8 1·787
110	124·7 (344·2)	3·60 1192·0 1·586	3·70 1201·3 1·597	3·76 1206·9 1·603	3·81 1212·4 1·610	3·87 1218·0 1·617	3·92 1223·5 1·624	3·98 1228·9 1·630	4·03 1234·3 1·636	4·08 1239·6 1·642	4·14 1245·0 1·648	4·19 1250·4 1·654	4·25 1255·6 1·660	4·30 1260·9 1·665	4·35 1266·1 1·670	4·45 1276·5 1·681	4·71 1302·0 1·707	4·97 1327·2 1·732	5·46 1377·2 1·778
120	134·7 (350·1)	3·34 1193·3 1·579	3·41 1199·3 1·587	3·46 1205·0 1·593	3 51 1210·6 1·600	3·57 1216·3 1·607	3·62 1221·8 1·614	3·67 1227·3 1·620	3·72 1232·8 1·626	3·77 1238·2 1·632	3·82 1243·6 1·638	3·87 1249·1 1·644	3·92 1254·4 1·650	3·97 1259·8 1·656	4·02 1265·0 1·661	4·11 1275·5 1·672	4·35 1301·1 1·698	4·59 1326·5 1·723	5·05 1376·7 1·769
130	144·7 (355·6)	3·12 1194·5 1·573	3·16 1197·2 1·577	3·21 1203·0 1·584	3·26 1208·8 1·590	3·30 1214·5 1·597	3·35 1220·2 1·604	3·40 1225·8 1·611	3·45 1231·3 1·617	3·50 1236·8 1·623	3·54 1242·3 1·629	3·59 1247·8 1·635	3·64 1253·2 1·641	3·68 1258·6 1·647	3·73 1263·9 1·653	3·82 1274·4 1·664	4·04 1300·3 1·690	4·27 1325·8 1·714	4·70 1376·2 1·760
140	154·7 (360·9)	2·93 1195·7 1·568	— — —	2·98 1201·0 1·574	3·03 1206·9 1·581	3·08 1212·7 1·588	3·12 1218·5 1·595	3·17 1224·2 1·602	3·21 1229·8 1·608	3·26 1235·4 1·615	3·30 1241·0 1·621	3·35 1246·5 1·627	3·39 1252·0 1·633	3·43 1257·4 1·639	3·48 1262·6 1·644	3·56 1273·4 1·656	3·78 1299·4 1·682	3·98 1325·1 1·706	4·39 1375·6 1·752

TABLE II. THE PROPERTIES OF SUPERHEATED STEAM—*continued*

VOLUME : CU. FT. PER LB.
TOTAL HEAT : BTU PER LB.
ENTROPY : BTU PER °F. ABS. PER LB.
} AT TEMPERATURES OF :—

PRESSURE PSI GAUGE	PSI ABS. (SAT. TEMP.)	Sat.	380°F.	390°F.	400°F.	410°F.	420°F.	430°F.	440°F.	450°F.	460°F.	470°F.	480°F.	490°F.	500°F.	550°F.	600°F.	650°F.	700°F.
160	174·7 (370·7)	2·61 1197·5 1·558	2·66 1203·3 1·564	2·70 1209·3 1·571	2·74 1215·1 1·578	2·78 1221·0 1·585	2·82 1226·8 1·592	2·86 1232·5 1·598	2·90 1238·1 1·604	2·94 1243·7 1·611	2·98 1249·3 1·616	3·02 1254·9 1·622	3·06 1260·3 1·628	3·10 1265·8 1·634	3·14 1271·2 1·640	3·33 1297·6 1·666	3·52 1323·5 1·692	3·70 1349·1 1·715	3·87 1374·8 1·738
180	194·7 (379·6)	2·35 1199·1 1·549	— — —	2·40 1205·7 1·556	2·44 1211·9 1·562	2·48 1217·8 1·569	2·51 1223·8 1·576	2·55 1229 6 1·582	2·59 1235·3 1·589	2·62 1241·0 1·595	2·66 1246·7 1·601	2·69 1252·4 1·607	2·73 1257·9 1·613	2·77 1263·4 1·620	2·80 1268·9 1·625	2·97 1295·8 1·652	3·14 1322·0 1·678	3·31 1347·9 1·702	3·47 1373·7 1·724
200	214·7 (387·7)	2·14 1200·4 1·541	— — —	2·15 1202·1 1·541	2·19 1208·4 1·549	2·22 1214·5 1·556	2·26 1220·7 1·563	2·29 1226·6 1·570	2·32 1232·4 1·576	2·36 1238·2 1·582	2·39 1244·1 1·589	2·42 1249·9 1·595	2·46 1255·5 1·601	2·49 1261·1 1·607	2·52 1266·7 1·613	2·68 1294·0 1·640	2·84 1320·5 1·666	2·99 1346·7 1·690	3·14 1372·6 1·713
220	234·7 (395·5)	1·96 1201·5 1·533	— — —	— — —	1·99 1204·7 1·536	2·02 1211·1 1·543	2·06 1217·4 1·550	2·09 1223·4 1·557	2·12 1229·5 1·563	2·15 1235·4 1·570	2·18 1241·4 1·577	2·21 1247·2 1·583	2·24 1253·0 1·589	2·27 1258·8 1·595	2·30 1264·4 1·601	2·45 1292·2 1·629	2·59 1319·0 1·656	2·73 1345·5 1·680	2·87 1371·5 1·703
240	254·7 (402·7)	1·81 1202·5 1·526	— — —	— — —	— — —	1·85 1207·5 1·532	1·88 1213·9 1·539	1·91 1220·2 1·545	1·94 1226·4 1·552	1·97 1232·5 1·559	2·00 1238·7 1·566	2·02 1244·6 1·572	2·05 1250·5 1·579	2·08 1256·4 1·585	2·11 1262·2 1·591	2·25 1290·4 1·620	2·38 1317·5 1·646	2·51 1344·3 1·670	2·64 1370·5 1·694
		Sat.	420°F.	430°F.	440°F.	450°F.	460°F.	470°F.	480°F.	490°F.	500°F.	520°F.	540°F.	560°F.	580°F.	600°F.	650F°.	700°F.	750°F.
260	274·7 (409·3)	1·68 1203·2 1·519	1·73 1210·3 1·527	1·76 1216·7 1·534	1·78 1223·1 1·541	1·81 1229·5 1·548	1·84 1235·8 1·555	1·87 1241·9 1·562	1·89 1248·0 1·569	1·92 1253·9 1·575	1·94 1259·9 1·581	2·00 1271·4 1·593	2·05 1283·0 1·605	2·10 1294·1 1·616	2·15 1305·1 1·626	2·20 1316·0 1·637	2·32 1343·0 1·662	2·44 1369·4 1·685	2·55 1395·6 1·707
280	294·7 (415·8)	1·57 1203·9 1·513	1·59 1206·5 1·516	1·62 1213·2 1·523	1·65 1219·9 1·531	1·67 1226·3 1·538	1·70 1232·8 1·545	1·73 1239·0 1·552	1·75 1245·4 1·559	1·78 1251·5 1·565	1·80 1257·6 1·572	1·85 1269·3 1·584	1·90 1281·0 1·596	1·95 1292·3 1·607	2·00 1303·4 1·618	2·04 1314·6 1·629	2·16 1341·7 1·653	2·27 1368·3 1·677	2·38 1394·6 1·699
300	314·7 (421·7)	1·47 1204·4 1·507	— — —	1·50 1209·5 1·513	1·53 1216·3 1·521	1·56 1223·2 1·528	1·58 1229·8 1·536	1·61 1236·2 1·543	1·63 1242·7 1·549	1·66 1248·9 1·556	1·68 1255·2 1·563	1·73 1267·2 1·575	1·77 1279·0 1·587	1·82 1290·5 1·598	1·86 1301·8 1·610	1·90 1313·1 1·620	1·99 1340·3 1·645	2·09 1367·1 1·669	2·19 1393·7 1·691
320	334·7 (427·5)	1·39 1204·9 1·502	— — —	1·40 1205·7 1·503	1·43 1212·8 1·511	1·45 1219·9 1·519	1·48 1226·7 1·526	1·50 1233·3 1·533	1·53 1239·9 1·540	1·55 1246·3 1·546	1·57 1252·8 1·554	1·61 1265·0 1·566	1·66 1277·0 1·578	1·70 1288·7 1·590	1·74 1300·1 1·601	1·79 1311·5 1·612	1·89 1339·1 1·638	1·99 1366·0 1·661	2·09 1392·8 1·683
340	354·7 (433·0)	1·31 1205·1 1·497	— — —	— — —	1·34 1209·3 1·502	1·36 1216·5 1·510	1·38 1223·6 1·517	1·41 1230·3 1·524	1·43 1237·1 1·531	1·45 1243·7 1·538	1·47 1250·3 1·545	1·52 1262·8 1·558	1·56 1274·9 1·571	1·60 1286·8 1·582	1·64 1298·5 1·594	1·68 1310·0 1·604	1·78 1337·8 1·630	1·87 1364·9 1·654	1·96 1391·8 1·676

TABLE II. THE PROPERTIES OF SUPERHEATED STEAM—*continued*

VOLUME : CU. FT. PER LB.
TOTAL HEAT : BTU PER LB.
ENTROPY : BTU PER °F. ABS. PER LB.
} AT TEMPERATURES OF :—

PRESSURE PSI GAUGE	PSI ABS. (SAT. TEMP.)	Sat.	450°F.	460°F.	470°F.	480°F.	490°F.	500°F.	520°F.	540°F.	560°F.	580°F.	600°F.	620°F.	640°F.	660°F.	680°F.	700°F.	750°F.
360	374·7 (438·3)	1·24 1205·4 1·492	1·28 1213·1 1·501	1·30 1220·2 1·508	1·32 1227·3 1·515	1·34 1234·2 1·523	1·37 1241·1 1·530	1·39 1247·8 1·537	1·43 1260·4 1·550	1·47 1272·8 1·563	1·51 1284·9 1·574	1·55 1296·8 1·586	1·58 1308·3 1·597	1·62 1319·6 1·608	1·66 1330·8 1·618	1·69 1341·9 1·628	1·73 1352·9 1·638	1·77 1363·8 1·647	1·86 1390·7 1·670
380	394·7 (443·3)	1·18 1205·5 1·487	1·20 1209·9 1·492	1·22 1217·3 1·500	1·25 1224·5 1·507	1·27 1231·3 1·515	1·29 1238·3 1·522	1·31 1245·2 1·529	1·35 1258·0 1·542	1·39 1270·7 1·555	1 42 1282·9 1·567	1·46 1295·0 1·579	1·50 1306·6 1·590	1·53 1318·1 1·601	1·57 1329·4 1·611	1·60 1340·5 1·621	1·64 1351·7 1·631	1·67 1362·6 1·640	1·76 1389·8 1·663
400	414·7 (448·1)	1·12 1205·6 1·482	1·13 1206·6 1·484	1·16 1214·2 1·491	1·18 1221·4 1·499	1·20 1228·5 1 507	1·22 1235·5 1·514	1 24 1242·5 1·522	1·28 1255·7 1·535	1·32 1268·5 1·548	1·35 1281·0 1·560	1·39 1293·1 1·572	1·42 1305·0 1·583	1·46 1316·5 1·594	1·49 1328·0 1·604	1·53 1339·1 1·615	1·56 1350·5 1·625	1·59 1361·5 1·634	1·67 1388·8 1·657
420	434·7 (452·8)	1·07 1205·6 1·478	— — —	1·09 1210·8 1·483	1·11 1218·2 1·491	1·13 1225·5 1·499	1·15 1232·6 1·507	1·17 1239·8 1·514	1·21 1253·2 1·528	1·24 1266·3 1·541	1·28 1279·0 1·553	1·31 1291·3 1·565	1·35 1303·3 1·577	1·38 1315·0 1·588	1·41 1326·5 1·598	1·45 1337·8 1·608	1·48 1349·2 1·618	1·51 1360·3 1·628	1·59 1387·8 1·651
450	464·7 (459·5)	1·00 1205·5 1·472	— — —	— — —	1·03 1213·3 1·479	1·05 1220·9 1·488	1·07 1228·2 1·495	1 09 1235·6 1·503	1·12 1249·5 1·517	1·16 1262·9 1·530	1·19 1275·9 1·544	1·22 1288·5 1·556	1·26 1300·8 1·568	1·29 1312·6 1·579	1·32 1324·4 1·589	1·35 1335·8 1·600	1·38 1347·3 1·609	1·41 1358·6 1·619	1·48 1386·3 1·643
		Sat.	480°F.	490°F.	500°F.	520°F.	540°F.	560°F.	580°F.	600°F.	620°F.	640°F.	660°F.	680°F.	700°F.	750°F.	800°F.	850°F.	900°F.
500	514·7 (470·0)	·902 1205·3 1·462	·930 1213·0 1·469	·949 1220·7 1·478	·966 1228·5 1·485	1·00 1243·1 1·500	1·03 1257·3 1·514	1·06 1270·8 1·528	1·09 1283·7 1·541	1·12 1296·4 1·553	1·15 1308·6 1·564	1·18 1320·7 1·575	1·21 1332·4 1·586	1·24 1344·1 1·596	1·26 1355·7 1·606	1·33 1383·9 1·630	1·40 1411·4 1·653	1·46 1438·8 1·674	1·53 1465·8 1·694
550	564·7 (479·7)	·820 1204·7 1·453	— — —	·845 1212·8 1·461	·862 1220·4 1·469	·894 1236·4 1·485	·924 1251·3 1·500	·955 1265·4 1·514	·984 1278·8 1·527	1·01 1292·0 1·539	1·04 1304·5 1·551	1·07 1316·9 1·562	1 09 1328·9 1·573	1·12 1340·8 1·584	1·14 1352·5 1·594	1·21 1381·3 1·618	1·27 1409·3 1·641	1·33 1436·9 1·662	1·38 1464·0 1·682
600	614·7 (488·8)	·750 1203·9 1·444	— — —	·763 1204·8 1·445	·780 1213·3 1·453	·810 1229·6 1·470	·839 1245·2 1·486	·868 1259·9 1·500	·895 1273·8 1·514	·921 1287·3 1·526	·947 1300·3 1·538	·972 1313·0 1·550	·997 1325·3 1·562	1 02 1337·5 1·572	1·05 1349·4 1·582	1·10 1378·6 1·607	1·16 1407·1 1·630	1·22 1435·0 1·652	1·27 1462·3 1·672
650	664·7 (497·3)	·692 1202·9 1·436	— — —	— — —	·705 1205·1 1·438	·735 1222·4 1·456	·765 1238·7 1·472	·791 1254·1 1·488	·818 1268·6 1·501	·843 1282·6 1·515	·867 1295·1 1·527	·891 1308·9 1·539	·914 1321·6 1·551	·938 1334·0 1·562	·960 1346·1 1·572	1·01 1375·9 1·597	1·07 1404·8 1·620	1·12 1433·0 1·642	1·17 1460·5 1·663
700	714·7 (505·4)	·641 1201·8 1·429	— — —	— — —	— — —	·672 1215·1 1·441	·699 1232·1 1·459	·726 1248·2 1·475	·751 1263·3 1·489	·775 1277·8 1·503	·798 1291·5 1·516	·821 1304·8 1·529	·843 1317·8 1·540	·865 1330·5 1·551	·885 1342·9 1·562	·939 1373·2 1·587	·988 1402·5 1·610	1·04 1430·9 1·633	1·08 1458·6 1·654

TABLE II. THE PROPERTIES OF SUPERHEATED STEAM—*continued*

VOLUME : CU. FT. PER LB.
TOTAL HEAT : BTU PER LB.
ENTROPY : BTU PER ° F. ABS. PER LB.
} AT TEMPERATURES OF :—

PRESSURE PSI GAUGE	PSI ABS. (SAT. TEMP.)	Sat.	530°F.	540°F.	550°F.	560°F.	570°F.	580°F.	590°F.	600°F.	620°F.	640°F.	660°F.	680°F.	700°F.	750°F.	800°F.	850°F.	900°F.
800	814·7 (520·3)	·557 1199·1 1·414	·577 1208·8 1·423	·591 1218·1 1·433	·604 1227·0 1·442	·617 1235·7 1·451	·628 1243·9 1·459	·640 1251·9 1·467	·652 1259·6 1·474	·663 1267·3 1·481	·684 1282·0 1·495	·706 1296·2 1·508	·726 1310·1 1·520	·745 1323·4 1·532	·766 1336·3 1·544	·813 1367·8 1·570	·857 1397·8 1·594	·900 1426·8 1·616	·944 1455·0 1·638
900	914·7 (533·9)	·490 1195·9 1·401	— — —	·504 1202·4 1·407	·517 1212·6 1·417	·529 1222·1 1·427	·541 1231·4 1·437	·554 1240·2 1·445	·565 1248·8 1·453	·576 1257·1 1·461	·596 1273·0 1·476	·617 1287·9 1·490	·636 1302·4 1·502	·655 1316·1 1·515	·673 1329·6 1·526	·716 1361·8 1·554	·758 1392·2 1·578	·796 1422·2 1·601	·835 1451·0 1·623
1000	1014·7 (546·4)	·437 1192·0 1·389	— — —	— — —	·446 1196·4 1·393	·459 1207·1 1·403	·473 1217·7 1·414	·485 1227·5 1·423	·496 1236·8 1·432	·508 1245·8 1·441	·527 1263·0 1·457	·547 1278·9 1·471	·565 1294·2 1·485	·582 1308·6 1·498	·600 1322·6 1·510	·640 1355·8 1·538	·678 1386·9 1·564	·714 1417·6 1·588	·749 1447·2 1·609
1100	1114·7 (557·9)	·394 1187·9 1·377	— — —	— — —	— — —	·401 1190·7 1·380	·413 1202·3 1·391	·424 1213·3 1·402	·435 1223·6 1·412	·447 1233·4 1·421	·467 1252·0 1·439	·486 1269·1 1·454	·503 1285·3 1·469	·521 1300·6 1·482	·537 1315·3 1·494	·575 1349·9 1·524	·610 1382·0 1·550	·646 1413·2 1·574	·677 1443·3 1·596
1200	1214·7 (568·8)	·357 1183·4 1·366	— — —	— — —	— — —	— — —	·362 1185·1 1·368	·373 1197·3 1·380	·385 1209·3 1·390	·396 1220·3 1·401	·416 1240·2 1·420	·435 1258·6 1·437	·452 1275·9 1·453	·468 1292·2 1·467	·485 1307·6 1·480	·522 1343·9 1·511	·555 1377·4 1·538	·588 1409·0 1·562	·618 1439·5 1·585
		Sat.	**600°F.**	**620°F.**	**640°F.**	**660°F.**	**680°F.**	**700°F.**	**720°F.**	**740°F.**	**760°F.**	**780°F.**	**800°F.**	**820°F.**	**840°F.**	**860°F.**	**880°F.**	**900°F.**	**1000°F.**
1400	1414·7 (588·4)	·298 1173·3 1·345	·313 1188·7 1·359	·333 1214 1 1 383	·353 1236·4 1·404	·370 1256·6 1·422	·387 1274·9 1·438	·402 1292·0 1 453	·417 1308·6 1·466	·430 1323·8 1·480	·442 1338·7 1 491	·455 1353·0 1·503	·467 1367·0 1·514	·478 1380·4 1·524	·490 1393·6 1·535	·501 1406·6 1·545	·511 1419·2 1·555	·522 1431·7 1·564	·574 1491·2 1·606
1600	1614·7 (606·1)	·254 1162·0 1·324	— — —	·270 1183·6 1·344	·290 1211·2 1·370	·307 1234·9 1·391	·324 1256·1 1·409	·340 1275·3 1·426	·353 1293·3 1·441	·366 1310·1 1·455	·378 1326·2 1·469	·389 1341·5 1·482	·400 1356·1 1·494	·410 1370·4 1·504	·421 1384·2 1·515	·431 1397·6 1·526	·441 1410·7 1·535	·451 1423·6 1·544	·497 1485·0 1·588
1800	1814·7 (622·1)	·215 1149·1 1·304	— — —	— — —	·239 1181·4 1·332	·258 1210·0 1·358	·274 1234·8 1·380	·289 1256·9 1·399	·303 1276·9 1·417	·314 1295·3 1·432	·326 1312·7 1·447	·337 1329·2 1·460	·348 1344·7 1·473	·358 1359·7 1·484	·368 1374·2 1·496	·377 1388·1 1·507	·386 1401·8 1·516	·395 1415·3 1·526	·438 1478·6 1·572
2000	2014·7 (636·8)	·186 1135·0 1·284	— — —	— — —	·193 1141·4 1·291	·214 1179·9 1·324	·232 1209·8 1·351	·246 1236·1 1·373	·260 1259·0 1·393	·273 1279·5 1·410	·284 1298·3 1·426	·295 1316·0 1·440	·306 1332·7 1·454	·316 1348·5 1·466	·325 1363·8 1·478	·333 1378·4 1·489	·342 1392·6 1·500	·350 1406·6 1·510	·391 1472·0 1·556
3000	3014·7 (696·1)	·0847 1012·7 1·155	— — —	— — —	— — —	— — —	— — —	— — —	— — —	·140 1174·4 1·293	·155 1208·7 1·322	·167 1237·3 1·344	·177 1262·2 1·364	·186 1284·7 1·382	·195 1305·2 1·397	·202 1324·3 1·413	·208 1342·4 1·426	·214 1359·7 1·438	·246 1435·3 1·492

TABLE III (Sect. 39). FLOW OF STEAM THROUGH PIPES

Approximate Weights in Pounds of Dry Saturated Steam per Minute that will Flow through 100 *ft. of various Sizes of Piping with a loss of One psi of Pressure:*

PRESSURE	DIAMETER OF PIPE IN INCHES											
PSI.G.	¾"	1"	1¼"	2"	2½"	3"	4"	5"	6"	8"	10"	12"
5	·53	1·17	3·50	7·4	13·0	20·7	42·7	75·0	118	247	430	717
15	·67	1·42	4·25	9·0	15·7	25·0	51·8	91·3	143	300	523	875
30	·80	1·67	5·16	10·8	19·1	30·3	62·7	110	173	362	632	1055
45	·92	1·97	5·83	12·4	21·7	34·7	70·0	126	198	413	722	1203
60	1·00	2·17	6·50	13·8	24·2	38·7	79·7	140	221	460	803	1340
80	1·13	2·42	7·16	15·5	27·2	43·3	89·2	157	247	515	900	1500
100	1·25	2·67	8·00	16·9	29·6	47·3	96·3	172	271	565	983	1645
120	1·37	2·92	8·85	18·6	32·7	52·0	108	188	297	620	1083	1800
150	1·48	3·17	9·50	20·0	35·3	56·0	116	204	322	672	1172	1950
200	1·67	3·67	10·83	22·9	40·3	64·2	133	233	367	767	1333	2230
250	1·87	4·00	12·00	25·3	44·6	71·0	147	258	407	847	1475	2470
300	2·00	4·33	13·00	27·2	47·6	76·0	157	275	433	905	1580	2630

(See also Sections 171 to 174)

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TABLE IV (Sect. 44). PER CENT. OF FLASH STEAM

INITIAL PRESSURE \ FINAL PRESSURE PSI. G.																								
FINAL PRESSURE PSI. G.	230	220	210	200	190	180	170	160	150	140	130	120	110	100	90	80	70	60	50	40	30	20	10	0
250 psi.g.	·91	1·37	1·86	2·35	2·86	3·37	3·90	4·45	5·02	5·63	6·24	6·87	7·55	8·25	8·98	9·77	10·62	11·55	12·52	13·64	14·93	16·40	18·24	20·76
240 "	·46	·93	1·41	1·91	2·42	2·93	3·47	4·02	4·59	5·19	5·81	6·45	7·13	7·83	8·56	9·35	10·21	11·15	12·12	13·24	14·53	16·01	17·85	20·37
230 "	—	·47	·96	1·46	1·97	2·48	3·02	3·57	4·15	4·75	5·38	6·01	6·70	7·40	8·13	8·93	9·80	10·73	11·71	12·82	14·12	15·60	17·45	19·99
220 "	—	—	·49	·99	1·51	2·02	2·56	3·12	3·69	4·30	4·93	5·56	6·25	6·96	7·69	8·49	9·36	10·30	11·28	12·41	13·70	15·19	17·05	19·59
210 "	—	—	—	·50	1·02	1·54	2·08	2·64	3·22	3·83	4·45	5·10	5·78	6·49	7·23	8·03	8·90	9·84	10·83	11·96	13·26	14·75	16·61	19·16
FINAL PRESSURE PSI. G.	190	180	170	160	150	140	130	120	110	100	90	80	70	60	50	40	35	30	25	20	15	10	5	0
200 psi.g.	·52	1·04	1·59	2·15	2·73	3·34	3·97	4·61	5·31	6·01	6·75	7·56	8·43	9·38	10·36	11·50	12·13	12·81	13·50	14·30	15·18	16·18	17·33	18·73
190 "	—	·52	1·07	1·64	2·21	2·83	3·46	4·11	4·81	5·51	6·26	7·07	7·94	8·90	9·89	11·03	11·65	12·32	13·03	13·83	14·71	15·71	16·88	18·28
180 "	—	—	·55	1·12	1·70	2·32	2·95	3·60	4·30	5·01	5·76	6·58	7·46	8·41	9·41	10·55	11·18	11·86	12·56	13·37	14·25	15·25	16·42	17·82
170 "	—	—	—	·57	1·15	1·77	2·41	3·06	3·77	4·48	5·24	6·05	6·93	7·89	8·89	10·04	10·67	11·36	12·06	12·87	13·75	14·75	15·92	17·34
160 "	—	—	—	—	·58	1·21	1·85	2·50	3·21	3·92	4·68	5·50	6·39	7·35	8·35	9·51	10·14	10·82	11·53	12·35	13·23	14·25	15·41	16·83
FINAL PRESSURE PSI. G.	140	130	120	110	100	90	85	80	75	70	65	60	55	50	45	40	35	30	25	20	15	10	5	0
150 psi.g.	·63	1·27	1·93	2·64	3·36	4·12	4·53	4·94	5·39	5·83	6·30	6·80	7·27	7·81	8·37	8·96	9·60	10·28	11·00	11·78	12·70	13·71	14·90	16·32
140 "	—	·65	1·31	2·02	2·74	3·51	3·92	4·34	4·79	5·23	5·70	6·20	6·68	7·22	7·78	8·37	9·02	9·71	10·41	11·25	12·13	13·15	14·33	15·76
130 "	—	—	·67	1·38	2·11	2·87	3·29	3·71	4·16	4·61	5·08	5·58	6·06	6·60	7·17	7·77	8·42	9·10	9·82	10·65	11·54	12·56	13·75	15·19
120 "	—	—	—	·72	1·45	2·22	2·64	3·06	3·52	3·96	4·44	4·94	5·42	5·96	6·54	7·14	7·79	8·48	9·20	10·03	10·92	11·95	13·15	14·59
110 "	—	—	—	—	·74	1·51	1·93	2·35	2·81	3·26	3·74	4·24	4·73	5·27	5·85	6·45	7·10	7·80	8·53	9·36	10·26	11·30	12·49	13·94
FINAL PRESSURE PSI. G.	95	90	85	80	75	70	65	60	55	50	45	40	35	30	25	20	18	16	14	12	10	8	6	0
100 psi.g.	·38	·78	1·20	1·62	2·09	2·54	3·02	3·52	4·02	4·56	5·14	5·75	6·40	7·10	7·83	8·67	9·02	9·38	9·78	10·18	10·61	11·08	11·56	13·27
95 "	—	·39	·82	1·24	1·71	2·16	2·64	3·15	3·64	4·19	4·77	5·38	6·03	6·74	7·47	8·31	8·66	9·03	9·42	9·82	10·25	10·72	11·21	12·92
90 "	—	—	·43	·85	1·32	1·77	2·25	2·76	3·26	3·80	4·39	5·00	5·66	6·36	7·10	7·94	8·28	8·66	9·05	9·45	9·88	10·36	10·84	12·56
85 "	—	—	—	·43	·89	1·35	1·83	2·34	2·84	3·39	3·97	4·59	5·24	5·95	6·69	7·53	7·88	8·26	8·65	9·05	9·49	9·96	10·45	12·17
80 "	—	—	—	—	·47	·92	1·41	1·92	2·42	2·97	3·56	4·17	4·83	5·54	6·28	7·13	7·48	7·85	8·24	8·65	9·09	9·56	10·05	11·77
FINAL PRESSURE PSI. G.	70	65	60	55	50	45	40	35	30	28	26	24	22	20	18	16	14	12	10	8	6	4	2	0
75 psi.g.	·46	·94	1·46	1·96	2·51	3·10	3·71	4·38	5·09	5·38	5·68	5·99	6·33	6·68	7·03	7·41	7·80	8·21	8·65	9·13	9·61	10·14	10·70	11·34
70 "	—	·49	1·01	1·51	2·06	2·65	3·27	3·94	4·65	4·94	5·24	5·56	5·90	6·24	6·60	6·97	7·37	7·78	8·22	8·69	9·18	9·72	10·28	10·92
65 "	—	—	·52	1·02	1·58	2·17	2·79	3·46	4·18	4·46	4·77	5·09	5·43	5·78	6·13	6·51	6·90	7·31	7·76	8·24	8·73	9·26	9·83	10·47
60 "	—	—	—	·51	1·06	1·66	2·28	2·95	3·67	3·96	4·26	4·58	4·93	5·28	5·63	6·01	6·41	6·82	7·26	7·74	8·24	8·77	9·34	9·98
55 "	—	—	—	—	·56	1·16	1·78	2·45	3·17	3·47	3·77	4·09	4·44	4·79	5·14	5·52	5·92	6·34	6·78	7·26	7·76	8·29	8·87	9·51

TABLE IV. PER CENT. OF FLASH STEAM—*continued*

Initial pressure \ Final pressure psi. g.	48	46	44	42	40	38	36	34	32	30	28	26	24	22	20	18	16	14	12	10	8	6	4	0
50 psi.g.	·24	·48	·72	·97	1·23	1·49	1·76	2·04	2·32	2·62	2·92	3·22	3·55	3·89	4·25	4·60	4·98	5·39	5·80	6·24	6·73	7·23	7·76	8·98
48 "	—	·24	·48	·73	·99	1·25	1·53	1·80	2·08	2·39	2·68	2·99	3·31	3·66	4·01	4·37	4·75	5·15	5·57	6·01	6·50	7·00	7·53	8·76
46 "	—	—	·24	·49	·75	1·01	1·29	1·57	1·84	2·15	2·45	2·75	3·08	3·42	3·78	4·14	4·52	4·92	5·34	5·78	6·27	6·77	7·30	8·53
44 "	—	—	—	·25	·51	·77	1·05	1·33	1·61	1·91	2·21	2·52	2·84	3·19	3·54	3·90	4·29	4·69	5·11	5·55	6·04	6·54	7·07	8·30
42 "	—	—	—	—	·26	·52	·80	1·08	1·36	1·67	1·96	2·27	2·60	2·94	3·30	3·66	4·04	4·44	4·86	5·31	5·80	6·30	6·84	8·06

Initial pressure \ Final pressure psi. g.	38	36	34	32	30	28	26	24	22	20	18	16	14	12	10	9	8	7	6	5	4	3	2	0
40 psi.g.	·26	·54	·82	1·10	1·41	1·71	2·01	2·34	2·69	3·04	3·41	3·79	4·19	4·61	5·06	5·30	5·54	5·80	6·05	6·31	6·59	6·87	7·17	7·82
38 "	—	·28	·56	·84	1·15	1·45	1·76	2·08	2·43	2·79	3·15	3·53	3·94	4·36	4·81	5·05	5·30	5·54	5·80	6·06	6·34	6·62	6·92	7·57
36 "	—	—	·28	·56	·87	1·17	1·48	1·81	2·15	2·51	2·87	3·26	3·66	4·08	4·53	4·78	5·02	5·28	5·52	5·79	6·07	6·36	6·65	7·30
34 "	—	—	—	·28	·59	·89	1·20	1·53	1·88	2·23	2·60	2·98	3·39	3·81	4·26	4·51	4·75	5·00	5·26	5·52	5·80	6·09	6·38	7·04
32 "	—	—	—	—	·31	·61	·92	1·25	1·60	1·96	2·32	2·71	3·11	3·54	3·99	4·23	4·48	4·73	4·98	5·25	5·53	5·82	6·11	6·77

Initial pressure \ Final pressure psi. g.	28	26	24	22	20	19	18	17	16	15	14	13	12	11	10	9	8	7	6	5	4	3	2	0
30 psi.g.	·30	·61	·94	1·29	1·65	1·83	2·01	2·20	2·40	2·60	2·81	3·01	3·23	3·46	3·68	3·93	4·17	4·43	4·68	4·94	5·23	5·52	5·81	6·47
28 "	—	·31	·64	·99	1·35	1·53	1·72	1·91	2·11	2·30	2·51	2·72	2·94	3·16	3·39	3·63	3·88	4·14	4·39	4·65	4·94	5·23	5·52	6·18
26 "	—	—	·33	·68	1·04	1·22	1·41	1·60	1·80	2·00	2·20	2·41	2·63	2·86	3·09	3·33	3·58	3·83	4·09	4·35	4·64	4·93	5·22	5·88
24 "	—	—	—	·35	·71	·89	1·08	1·27	1·47	1·67	1·88	2·09	2·30	2·53	2·76	3·01	3·25	3·51	3·77	4·03	4·31	4·61	4·91	5·56
22 "	—	—	—	—	·36	·54	·73	·92	1·12	1·32	1·53	1·74	1·96	2·19	2·41	2·66	2·91	3·16	3·42	3·68	3·97	4·26	2·56	5·22

Initial pressure \ Final pressure	19	18	17	16	15	14	13	12	11	10	9	8	7	6	5	4	3	2	1	0	1"	2"	3"	4"
20 psi.g.	·18	·37	·56	·76	·96	1·17	1·38	1·60	1·83	2·06	2·30	2·55	2·81	3·06	3·33	3·62	3·91	4·21	4·54	4·87	5·05	5·23	5·41	5·60
19 "	—	·19	·38	·58	·78	·99	1·20	1·42	1·65	1·88	2·13	2·37	2·63	2·89	3·15	3·44	3·73	4·04	4·37	4·70	4·88	5·05	5·24	5·42
18 "	—	—	·19	·39	·59	·80	1·01	1·23	1·46	1·69	1·94	2·19	2·44	2·70	2·97	3·25	3·55	3·85	4·18	4·51	4·69	4·87	5·05	5·24
17 "	—	—	—	·20	·40	·61	·82	1·04	1·27	1·50	1·75	2·00	2·26	2·51	2·78	3·06	3·36	3·66	3·99	4·33	4·51	4·69	4·87	5·05
16 "	—	—	—	—	·20	·41	·62	·84	1·07	1·30	1·55	1·80	2·06	2·31	2·58	2·87	3·16	3·47	3·80	4·13	4·31	4·49	4·67	4·86

Initial pressure \ Final pressure	14	13	12	11	10	9	8	7	6	5	4	3	2	1	0	1"	2"	3"	4"	5"	6"	7"	8"	9"
15 psi.g.	·21	·42	·64	·87	1·10	1·35	1·60	1·86	2·12	2·38	2·67	2·97	3·27	3·60	3·94	4·12	4·30	4·48	4·67	4·85	5·04	5·25	5·47	5·69
14 "	—	·21	·43	·66	·89	1·14	1·39	1·65	1·91	2·17	2·46	2·76	3·06	3·39	3·73	3·91	4·09	4·27	4·46	4·65	4·84	5·04	5·26	5·49
13 "	—	—	·22	·45	·68	·93	1·18	1·44	1·70	1·97	2·25	2·55	2·86	3·19	3·52	3·70	3·89	4·07	4·25	4·44	4·63	4·84	5·06	5·28
12 "	—	—	—	·23	·46	·71	·96	1·22	1·48	1·75	2·04	2·33	2·64	2·97	3·31	3·49	3·67	3·85	4·04	4·23	4·42	4·63	4·85	5·07
11 "	—	—	—	—	·23	·48	·73	·99	1·25	1·52	1·81	2·10	2·41	2·74	3·16	3·26	3·44	3·62	3·82	4·00	4·19	4·40	4·62	4·83

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TABLE IV. PER CENT. OF FLASH STEAM—*continued*

FINAL PRESSURE / INITIAL PRESSURE	9	8	7	6	5	4	3	2	1	0	1"	2"	3"	4"	5"	6"	7"	8"	9"	10"	11"	12"	13"	14"
10 psi.g.	·25	·50	·76	1·02	1·29	1·58	1·88	2·18	2·51	2·86	3·04	3·22	3·40	3·59	3·78	3·97	4·18	4·40	4·62	4·85	5·09	5·33	5·59	5·87
9 „	—	·25	·51	·76	1·02	1·33	1·63	1·93	2·27	2·61	2·79	2·97	3·15	3·34	3·53	3·72	3·93	4·15	4·38	4·61	4·84	5·09	5·35	5·63
8 „	—	—	·26	·52	·79	1·08	1·38	1·69	2·02	2·37	2·54	2·72	2·91	3·10	3·29	3·48	3·69	3·91	4·14	4·36	4·60	4·85	5·11	5·39
7 „	—	—	—	·26	·53	·82	1·12	1·43	1·76	2·11	2·28	2·47	2·65	2·84	3·03	3·22	3·43	3·65	3·88	4·11	4·34	4·60	4·85	5·14
6 „	—	—	—	—	·27	·56	·86	1·17	1·50	1·85	2·03	2·21	2·39	2·58	2·77	2·97	3·18	3·40	3·63	3·86	4·09	4·34	4·60	4·88
FINAL PRESSURE / INITIAL PRESSURE	**4**	**3**	**2**	**1**	**0**	**1"**	**2"**	**3"**	**4"**	**5"**	**6"**	**7"**	**8"**	**9"**	**10"**	**11"**	**12"**	**13"**	**14"**	**15"**	**16"**	**17"**	**18"**	**19"**
5 psi.g.	·30	·59	·90	1·23	1·58	1·76	1·94	2·12	2·32	2·51	2·70	2·91	3·13	3·36	3·59	3·83	4·08	4·33	4·62	4·91	5·21	5·52	5·86	6·22
4 „	—	·30	·61	·94	1·29	1·47	1·65	1·84	2·03	2·22	2·41	2·62	2·85	3·08	3·31	3·55	3·79	4·05	4·34	4·63	4·92	5·24	5·58	5·94
3 „	—	—	·31	·64	·99	1·17	1·36	1·54	1·73	1·92	2·12	2·33	2·55	2·78	3·01	3·25	3·50	3·76	4·04	4·34	4·63	4·94	5·29	5·65
2 „	—	—	—	·33	·68	·86	1·02	1·23	1·43	1·62	1·81	2·02	2·24	2·47	2·70	2·94	3·19	3·45	3·74	4·04	4·33	4·64	4·98	5·35
1 „	—	—	—	—	·35	·54	·72	·90	1·10	1·29	1·48	1·70	1·92	2·15	2·38	2·62	2·87	3·13	3·42	3·71	4·01	4·32	4·66	5·03
Atmos.	—	—	—	—	—	·19	·37	·55	·75	·94	1·14	1·35	1·57	1·80	2·03	2·27	2·52	2·78	3·07	3·37	3·67	3·98	4·32	4·69
FINAL PRESSURE IN. VAC. / INITIAL PRESSURE	**2"**	**3"**	**4"**	**5"**	**6"**	**7"**	**8"**	**9"**	**10"**	**11"**	**12"**	**13"**	**14"**	**15"**	**16"**	**17"**	**18"**	**19"**	**20"**	**21"**	**22"**	**23"**	**24"**	**25"**
1" vac.	·19	·37	·56	·76	·95	1·16	1·39	1·62	1·85	2·09	2·34	2·60	2·89	3·19	3·48	3·80	4·14	4·52	4·90	5·33	5·80	6·30	6·87	8·37
2" „	—	·18	·38	·59	·77	·98	1·20	1·44	1·67	1·91	2·16	2·42	2·71	3·01	3·30	3·62	3·96	4·34	4·72	5·15	5·62	6·12	6·70	8·19
3" „	—	—	·19	·39	·58	·80	1·02	1·25	1·49	1·73	1·98	2·24	2·53	2·82	3·12	3·44	3·78	4·15	4·54	4·97	5·44	5·95	6·52	8·01
4" „	—	—	—	·19	·39	·60	·83	1·06	1·29	1·53	1·78	2·04	2·33	2·63	2·93	3·25	3·59	3·96	4·35	4·78	5·25	5·76	6·33	7·83
5" „	—	—	—	—	·19	·41	·63	·87	1·10	1·34	1·59	1·85	2·14	2·44	2·74	3·06	3·40	3·77	4·16	4·59	5·06	5·57	6·15	7·64
FINAL PRESSURE IN. VAC. / INITIAL PRESSURE	**7"**	**8"**	**9"**	**10"**	**11"**	**12"**	**13"**	**14"**	**15"**	**16"**	**17"**	**18"**	**19"**	**20"**	**21"**	**22"**	**23"**	**24"**	**25"**	**26"**	**27"**	**28"**		
6" vac.	·21	·44	·67	·91	1·15	1·40	1·66	1·95	2·25	2·55	2·86	3·21	3·58	3·97	4·40	4·87	5·38	5·96	6·64	7·42	8·38	9·66		
7" „	—	·22	·46	·69	·93	1·19	1·45	1·74	2·04	2·34	2·65	3·00	3·37	3·76	4·19	4·67	5·18	5·75	6·43	7·21	8·17	9·45		
8" „	—	—	·23	·47	·71	·96	1·23	1·51	1·82	2·11	2·43	2·78	3·15	3·54	3·97	4·45	4·97	5·53	6·22	7·00	7·96	9·24		
9" „	—	—	—	·23	·48	·73	·99	1·28	1·58	1·88	2·20	2·55	2·92	3·31	3·74	4·22	4·73	5·31	5·99	6·78	7·74	9·02		
10" „	—	—	—	—	·24	·50	·76	1·02	1·35	1·65	1·97	2·32	2·69	3·08	3·52	3·99	4·50	5·08	5·77	6·55	7·51	8·80		
FINAL PRESSURE IN. VAC. / INITIAL PRESSURE	**12"**	**13"**	**14"**	**15"**	**16"**	**17"**	**18"**	**19"**	**20"**	**21"**	**22"**	**23"**	**24"**	**25"**	**26"**	**27"**	**28"**							
11" vac.	·25	·52	·81	1·11	1·41	1·73	2·08	2·45	2·85	3·28	3·75	4·27	4·84	5·53	6·32	7·28	8·57							
12" „	—	·26	·56	·86	1·16	1·48	1·83	2·20	2·60	3·03	3·50	4·02	4·60	5·29	6·07	7·04	8·33							
13" „	—	—	·29	·60	·90	1·22	1·56	1·94	2·34	2·77	3·24	3·76	4·34	5·03	5·82	6·79	8·07							
14" „	—	—	—	·30	·60	·92	1·27	1·65	2·05	2·48	2·95	3·47	4·05	4·74	5·53	6·50	7·79							
15" „	—	—	—	—	·30	·62	·97	1·35	1·75	2·18	2·66	3·18	3·76	4·45	5·24	6·21	7·50							

TABLE IV. PER CENT. OF FLASH STEAM—*continued*

INITIAL PRESSURE \ FINAL PRESSURE IN. VAC.	17"	18"	19"	20"	21"	22"	23"	24"	25"	26"	27"	28"
16" vac.	·32	·67	1·05	1·45	1·88	2·36	2·88	3·46	4·16	4·95	5·82	7·21
17" „	—	·35	·73	1·13	1·56	2·04	2·56	3·15	3·84	4·63	5·61	6·91
18" „	—	—	·38	·78	1·21	1·70	2·22	2·80	3·50	4·29	5·27	6·57
19" „	—	—	—	·40	·84	1·32	1·84	2·43	3·12	3·92	4·90	6·20
20" „	—	—	—	—	·44	·92	1·44	2·03	2·73	3·53	4·51	5·82

INITIAL PRESSURE \ FINAL PRESSURE IN. VAC.	22"	23"	24"	25"	26"	27"	28"
21" vac	·49	1·01	1·60	2·30	3·10	4·08	5·39
22" „	—	·52	1·11	1·82	2·62	3·61	4·92
23" „	—	—	·59	1·30	2·10	3·09	4·41
24" „	—	—	—	·71	1·52	2·51	3·83
25" „	—	—	—	—	·81	1·81	3·13

TABLE V (Sect. 44). FLASH FOR 1 LB. CONDENSATE COLLECTED AT 212° F.

PRESSURE PSI.G.	WEIGHT OF INPUT STEAM DRY SATURATED	LATENT HEAT GIVEN UP IN PLANT	WEIGHT OF FLASH	HEAT IN FLASH
	LB.	BTU.	LB.	BTU.
5	1·0161	976·2	·0161	18·5
10	1·0294	981·0	·0294	33·9
15	1·0410	984·8	·0410	47·2
20	1·0512	988·2	·0512	58·9
25	1·0607	991·3	·0607	69·8
30	1·0692	994·0	·0692	79·6
35	1·0772	996·3	·0772	88·9
40	1·0848	998·5	·0848	97·6
45	1·0918	1,000·3	·0918	105·7
50	1·0987	1,002·2	·0987	113·5
60	1·1109	1,005·7	·1109	127·6
70	1·1226	1,009·0	·1226	141·1
80	1·1334	1,011·8	·1334	153·5
90	1·1436	1,014·4	·1436	165·3
100	1·1530	1,016·5	·1530	176·1

TABLE VI (Sect. 48). SUPERHEAT DUE TO WIREDRAWING

INITIAL SATURATED STEAM PRESSURE PSI.G.	APPROXIMATE SUPERHEAT IN °F. WHEN INITIAL STEAM BLOWN DOWN TO PRESSURES PSI.G. :—														
	2	4	6	8	10	15	20	25	30	35	40	45	50	75	100
5	6	2													
10	14	10	6	3											
15	22	18	14	11	8										
20	28	24	20	17	14	6									
25	34	30	26	23	20	12	6								
30	38	34	30	27	24	16	10	4							
35	42	38	34	31	28	20	14	8	4						
40	46	42	38	35	32	24	18	12	8	4					
45	48	44	40	37	34	26	20	14	10	6	2				
50	52	48	44	41	38	30	24	18	14	10	6	4			
60	58	54	50	47	44	36	30	24	20	16	12	10	6		
70	62	58	54	51	48	40	34	28	24	20	16	14	10		
80	66	62	58	57	52	44	38	32	28	24	20	18	14	2	
90	70	66	62	59	56	48	42	36	32	28	24	22	18	6	
100	74	70	66	63	60	52	46	40	36	32	28	26	22	10	
120	78	74	70	67	64	56	50	44	40	36	32	30	26	14	4
140	84	80	76	73	70	62	56	50	46	42	38	36	32	20	10
160	87	83	79	76	73	65	59	53	49	45	41	39	35	23	13
180	90	86	82	79	76	68	62	56	52	48	44	42	38	26	16
200	92	88	84	81	78	70	64	58	54	50	46	44	40	28	18

TABLE VII (Sect. 52). ENERGY EQUIVALENTS

	MECHANICAL	THERMAL	
	FOOT-POUNDS	BTU	C.H.U.
Mechanical equivalent of heat	778 1,400	1·0 1·8	·56 1·0
Foot-ton	2,240	2·88	1·6
Foot-pound	1	·00129	·00072

TABLE VIII (Sect. 53). POWER EQUIVALENTS

	MECHANICAL			THERMAL		ELECTRICAL
	FT. LB./SEC.	FT. LB./MIN.	FT. LB./HR.	BTU/HR.	C.H.U./HR.	WATTS
1 horse-power	550	33,000	1,980,000	2,545	1,414	746
1 kilowatt ..	738	44,250	2,655,000	3,415	1,897	1,000

TABLE IX (Sect. 104). APPROXIMATE TURBINE EFFICIENCY RATIOS

(Assuming appropriate Superheat and appropriate Speed)

PRESSURE DROP \ HORSE POWER	250	500	750	1,000	2,000	3,000	4,000	5,000	7,500	10,000
1,000 psi.a. to 400	—	—	—	—	59	62	64	65	67	68
400 psi.a. to 100	47	54	57	59	64	67	68	69	71	72
100 psi.a. to 20	61	65	67	68	71	73	74	75	76	76
20 psi.a. to 28″ vac.	64	68	70	71	74	76	78	79	80	80

(This table must not be used without reading the explanation in the text.)

TABLE X (Sect. 104). IDEAL HORSE POWER AND APPROXIMATE EFFICIENCY RATIOS FOR ENGINES AND TURBINES USING 10,000 LB. SATURATED STEAM PER HOUR

INLET PRESS. SAT. PSI.G.	EXHAUST											
	VACUUM					GAUGE PRESSURE						
	20"	15"	10"	5"	ATM.	5	10	20	40	60	80	100
100 HP	855	760	690	635	590	475	455	355	230			
E	53	58	60	62	65	64	61	60	56	—	—	—
T	67	62	60	57	53	51	48	46	37	—	—	—
125 HP	910	815	745	680	650	570	520	420	290			
E	53	58	60	62	64	67	64	63	60	—	—	—
T	64	60	59	57	53	52	48	47	38	—	—	—
150 HP	955	860	810	735	695	615	560	465	340	250		
E	53	57	58	62	63	66	66	66	62	58	—	—
T	63	59	57	57	53	53	50	48	41	32	—	—
175 HP	985	895	852	775	735	655	595	500	380	290	215	
E	53	57	58	61	62	66	67	67	62	62	60	—
T	62	59	56	56	53	53	50	48	43	36	—	—
200 HP	1,020	930	885	810	770	690	635	540	420	330	255	195
E	53	56	57	60	61	64	66	67	65	62	61	60
T	62	59	56	56	53	53	50	49	45	39	35	—
225 HP	1,050	960	910	840	800	725	665	575	450	360	290	230
E	52	55	56	59	60	63	65	66	67	63	62	60
T	62	58	56	56	54	53	50	49	45	39	36	—
250 HP	1,075	985	940	870	825	750	700	610	485	395	325	265
E	—	—	—	—	59	62	63	65	67	64	61	60
T	62	58	56	56	54	53	50	49	46	61	37	—
300 HP	1,120	1,030	980	910	870	795	740	650	530	445	375	315
E	—	—	—	—	—	—	62	64	67	64	60	60
T	60	58	56	56	54	54	52	50	46	41	37	—
350 HP	1,150	1,065	1,015	945	910	835	780	690	575	485	415	360
E	—	—	—	—	—	—	62	64	66	64	60	60
T	60	59	57	56	54	54	53	50	46	42	38	—
400 HP	1,190	1,100	1,055	980	945	870	815	730	615	525	460	405
E	—	—	—	—	—	—	—	63	65	63	59	58
T	60	59	57	56	54	54	52	50	46	44	39	35

TABLE XI (Sect. 114). COMPARATIVE POWER GENERATING EFFICIENCY

	GOOD CONDENSING POWER PLANT	BAD SMALL COMBINED POWER AND HEATING PLANT	GOOD LARGE COMBINED POWER AND HEATING PLANT
Theoretical process efficiency	45%	100%	100%
Boiler efficiency	85%	50%	83%
Machine loss	5%	30%	10%
Coal consumption	·99 lb./kWh	·86 lb./kWh	·39 lb./kWh
Generating efficiency ..	28·7%	33%	73%
Overall efficiency		41%	81%

TABLE XII (Sect. 118). APPROXIMATE COAL CONSUMPTION OF POWER GENERATION

	APPROX. LB. COAL PER KWH
Back pressure—large good	·4
Back pressure—small bad	·9
Public supply—base load	1·0
Public supply—average	1·4
Public supply—peaks	2·7
Large industrial condensing engine ..	3·3
Non-expansive non-condensing pump ..	14·0

TABLE XIII (Sect. 120). BACK-PRESSURE POWER COSTS

	WINTER 1943–44		WINTER 1957–58
Size of plant—kW	350	4,000	9,000
Hours per week	70	140	130
Coal consumption—lb./kWh ..	·6	·4	·4
Coal cost	·15	·10	·24
Overheads and depreciation ..	·26	·09	·12
Maintenance	·08	·03	·04
Labour	·12	·03	·04
Miscellaneous	·02	·02	·01
	·63 d./kWh	·27 d./kWh	·45 d./kWh

TABLE XIV (Sect. 120). APPROXIMATE BACK PRESSURE POWER HEAT CONSUMPTION

SIZE OF SET KW	BTU TAKEN FROM STEAM PER KWH
3,000 and over	3,415 × 1·1
1,000 to 3,000	3,415 × 1·125
250 to 1,000	3,415 × 1·15
100 to 250	3,415 × 1·175
50 to 100	3,415 × 1·2
Less than 50	3,415 × 1·25

TABLE XV (Sect. 135). EXHAUST STEAM ACCUMULATORS

NET PRESSURE DROP IN ACCUMULATOR PSI.G.	GROSS PRESSURE DROP OVER PLANT PSI (HYDROSTATIC HEAD ·5 PSI)	LB. WATER PER LB. STEAM STORED	WATER CAPACITY NEEDED TO STORE 1,000 LB. STEAM CU. FT.	EQUIVALENT NO. OF LANCS. BOILER SHELLS (30′ × 8′ 6″)
1 to 0	2	340	5,460	6
2 to 0	3	174	2,790	3
3 to 0	4	120	1,930	2
4 to 0	5	92	1,475	$1\frac{2}{3}$
5 to 0	6	75	1,205	$1\frac{1}{3}$

TABLE XVI (Sect. 160). FLUID LOSS BY LEAKAGE

DIAMETER OF HOLE	STEAM—LB./HOUR		WATER—GALLS./HOUR		CU. FT. FREE AIR/MIN. 80 PSI
	100 PSI	300 PSI	20 PSI	100 PSI	
$\frac{1}{16}$″	14	33	20	45	4
$\frac{1}{8}$″	56	132	80	180	16
$\frac{3}{16}$″	126	297	180	405	36
$\frac{1}{4}$″	224	528	320	720	64

TABLE XVII (Sect. 160). COAL LOSS BY STEAM LEAKAGE

DIAMETER OF HOLE	TONS OF COAL PER ANNUM			
	100 PSI BOILER EFFICIENCY 65%		300 PSI BOILER EFFICIENCY 75%	
	48 HR. WEEK	144 HR. WEEK	48 HR. WEEK	144 HR. WEEK
$\frac{1}{16}$″	2	7	5	14
$\frac{1}{8}$″	9	28	19	56
$\frac{3}{16}$″	21	62	42	126
$\frac{1}{4}$″	37	111	74	223

TABLE XVIII (Sect. 160). STEEL STEAM PIPES

Nominal Pipe Diameter Inches	B.S.I. Table	Maximum Steam Pressure psi.	Flange					Weight	
			Diameter Inches	Thickness Inches	Diameter of Bolt Circle Inches	Number of Bolts	Size of Bolts Inch	Per Foot lb.	Pair of Flanges lb.
1/2	D	50	3 3/4	3/16	2 5/8	4	1/2	1·0	1·3
1/2	E	100	3 3/4	1/4	2 5/8	4	1/2	1·0	1·6
1/2	F	150	3 3/4	3/8	2 5/8	4	1/2	1·0	2·4
1/2	H	250	4 1/2	1/2	3 1/4	4	5/8	1·0	4·5
1/2	J	350	4 1/2	5/8	3 1/4	4	5/8	1·0	5·6
1/2	K	450	4 1/2	3/4	3 1/4	4	5/8	1·0	6·7
1/2	R	600	4 1/2	3/4	3 1/4	4	5/8	1·1	6·7
3/4	D	50	4	3/16	2 7/8	4	1/2	1·4	1·4
3/4	E	100	4	1/4	2 7/8	4	1/2	1·4	1·8
3/4	F	150	4	3/8	2 7/8	4	1/2	1·4	2·7
3/4	H	250	4 1/2	1/2	3 1/4	4	5/8	1·4	4·5
3/4	J	350	4 1/2	5/8	3 1/4	4	5/8	1·4	5·6
3/4	K	450	4 1/2	3/4	3 1/4	4	5/8	1·4	6·7
3/4	R	600	4 1/2	3/4	3 1/4	4	5/8	1·4	6·7
1	D	50	4 1/2	3/16	3 1/4	4	1/2	2·0	1·8
1	E	100	4 1/2	9/32	3 1/4	4	1/2	2·0	2·6
1	F	150	4 3/4	3/8	3 7/16	4	5/8	2·0	3·7
1	H	250	4 3/4	9/16	3 7/16	4	5/8	2·0	5·5
1	J	350	4 3/4	3/4	3 7/16	4	5/8	2·0	7·3
1	K	450	5	7/8	3 3/4	4	5/8	2·0	9·4
1	R	600	5	7/8	3 3/4	4	5/8	2·0	9·4
1 1/2	D	50	5 1/4	1/4	3 7/8	4	1/2	3·5	3·1
1 1/2	E	100	5 1/4	11/32	3 7/8	4	1/2	3·5	4·1
1 1/2	F	150	5 1/2	1/2	4 1/8	4	5/8	3·5	6·3
1 1/2	H	250	5 1/2	11/16	4 1/8	4	5/8	3·5	8·6
1 1/2	J	350	5 1/2	7/8	4 1/8	4	5/8	3·5	10·8
1 1/2	K	450	6	1	4 1/2	4	3/4	3·5	15·1
1 1/2	R	600	6	1	4 1/2	4	3/4	3·5	15·1
2	D	50	6	5/16	4 1/2	4	5/8	4·5	4·7
2	E	100	6	3/8	4 1/2	4	5/8	4·5	5·5
2	F	150	6 1/2	5/8	5	4	5/8	4·5	10·7
2	H	250	6 1/2	3/4	5	4	5/8	4·5	12·7
2	J	350	6 1/2	1	5	4	3/4	4·5	17·1
2	K	450	6 1/2	1	5	8	5/8	4·5	17·1
2	R	600	6 1/2	1	5	8	5/8	4·9	17·1
2 1/2	D	50	6 1/2	5/16	5	4	5/8	6·3	5·2
2 1/2	E	100	6 1/2	11/32	5	4	5/8	6·3	6·5
2 1/2	F	150	7 1/4	5/8	5 3/4	8	5/8	6·3	12·7
2 1/2	H	250	7 1/4	3/4	5 3/4	8	5/8	6·3	15·3

TABLE XVIII. STEEL STEAM PIPES—*continued*

NOMINAL PIPE DIAMETER INCHES	B.S.I. TABLE	MAXIMUM STEAM PRESSURE PSI	FLANGE					WEIGHT	
			DIAMETER INCHES	THICKNESS INCHES	DIAMETER OF BOLT CIRCLE INCHES	NUMBER OF BOLTS	SIZE OF BOLTS INCH	PER FOOT LB.	PAIR OF FLANGES LB.
2½	J	350	7½	1	5¾	8	¾	6·3	20·3
2½	K	450	7½	1⅛	5¾	8	¾	6·3	22·8
2½	R	600	7½	1⅛	5¾	8	¾	6·9	22·8
3	D	50	7½	⅜	5¾	4	⅝	7·5	7·4
3	E	100	7½	7/16	5¾	4	⅝	7·5	8·6
3	F	150	8	⅝	6½	8	⅝	7·5	15·1
3	H	250	8	⅞	6½	8	⅝	7·5	21·0
3	J	350	8	1¼	6½	8	¾	7·5	30·3
3	K	450	8	1¼	6½	8	¾	7·5	30·3
3	R	600	8	1¼	6½	8	¾	8·7	30·3
4	D	50	8½	⅜	7	4	⅝	9·7	9·5
4	E	100	8½	½	7	8	⅝	9·7	12·4
4	F	150	9	¾	7½	8	⅝	9·7	21·3
4	H	250	9	1	7½	8	⅝	9·7	28·5
4	J	350	9	1⅜	7½	8	¾	11·4	39·2
4	K	450	9½	1⅜	7¾	8	⅞	11·4	45
4	R	600	9½	1⅜	7¾	8	⅞	14·0	45
5	D	50	10	½	8¼	8	⅝	12·0	16·6
5	E	100	10	9/16	8¼	8	⅝	12·0	18·7
5	F	150	11	⅞	9¼	8	¾	12·0	37
5	H	250	11	1⅛	9¼	8	¾	14·0	47
5	J	350	11	1½	9¼	8	⅞	14·0	63·5
5	K	450	11	1⅝	9¼	12	⅞	15·7	69
5	R	600	11	1⅝	9¼	12	⅞	19	69
6	D	50	11	½	9¼	8	⅝	14·3	18·8
6	E	100	11	11/16	9¼	8	¾	14·3	25·5
6	F	150	12	⅞	10¼	12	¾	14·3	41
6	H	250	12	1⅛	10¼	12	¾	16·7	53
6	J	350	12	1½	10¼	12	⅞	16·7	71
6	K	450	12	1⅝	10¼	12	⅞	20·7	77
6	R	600	12	1¾	10¼	12	⅞	24·6	83
7	D	50	12	½	0¼	8	⅝	19·4	21
7	E	100	12	¾	10¼	8	¾	19·4	31
7	F	150	13½	⅞	11½	12	¾	19·4	49
7	H	250	13½	1¼	11½	12	¾	19·4	70
7	J	350	13½	1⅝	11½	12	⅞	21·7	91
7	K	450	13½	1¾	11½	12	1	26·3	103
7	R	600	13½	1⅞	11½	12	1	30·8	110
8	D	50	13½	⅝	11½	8	⅝	22·0	24·8
8	E	100	13½	¾	11½	8	¾	22·0	36·5
8	F	150	14½	1	12¾	12	¾	22·0	64
8	H	250	14½	1¼	12¾	12	¾	22·0	81

TABLE XVIII. STEEL STEAM PIPES—*continued*

NOMINAL PIPE DIAMETER INCHES	B.S.I. TABLE	MAXIMUM STEAM PRESSURE PSI.	FLANGE					WEIGHT	
			DIAMETER INCHES	THICKNESS INCHES	DIAMETER OF BOLT CIRCLE INCHES	NUMBER OF BOLTS	SIZE OF BOLTS INCH	PER FOOT LB.	PAIR OF FLANGES LB.
8	J	350	14½	1⅝	12¾	12	⅞	27·3	105
8	K	450	14½	1⅞	12¼	12	1	32·6	121
8	R	600	14½	2	12¾	12	1	40·2	129
9	D	50	14½	⅝	12¾	8	⅝	23·4	35·4
9	E	100	14½	13/16	12¾	12	¾	23·4	45·5
9	F	150	16	1	14	12	⅞	23·4	74
9	H	250	16	1⅜	14	12	⅞	23·4	106
9	J	350	16	1¾	14	12	1	29·0	135
9	K	450	16	2	14	16	1	37·3	153
9	R	600	16	2⅛	14	16	1	45·4	164
10	D	50	16	⅝	14	8	¾	30·7	43
10	E	100	16	⅞	14	12	¾	30·7	60
10	F	150	17	1	15	12	⅞	30·7	84
10	H	250	17	1⅜	15	12	⅞	30·7	114
10	J	350	17	1⅞	15	12	1	37·3	156
10	K	450	17	2	15	16	1	47·0	166
10	R	600	17	2¼	15¼	16	1	59·7	187
12	D	50	18	⅝	16	12	¾	40·7	50
12	E	100	18	1	16	12	⅞	40·7	78
12	F	150	19¼	1⅛	17¼	16	⅞	40·7	112
12	H	250	19¼	1½	17¼	16	⅞	40·7	149
12	J	350	19¼	2	17¼	16	1	48·6	199
12	K	450	19¼	2¼	17	16	1⅛	60·3	224
12	R	600	20	2⅝	18	16	1⅛	79·3	282
15	D	50	21¾	¾	19½	12	⅞	62·6	76
15	E	100	21¾	1	19½	12	⅞	62·6	102
15	F	160	22¾	1¼	20½	16	1	62·6	153
15	H	250	22¾	1¾	20½	16	1	62·6	214
15	J	350	22¾	2⅛	20½	16	1⅛	77·8	260
15	K	450	23¾	2½	21¼	20	1¼	97·7	356
15	R	600	24	2⅞	21¾	20	1¼	127	424
18	D	50	25¼	⅞	23	12	⅞	50·1	113
18	E	100	25¼	1⅛	23	16	⅞	56·3	145
18	F	150	26½	1⅜	24	20	1⅛	68·5	218
18	H	250	26½	1⅞	24	20	1⅛	98·8	297
18	J	350	26½	2⅜	24	20	1¼	128·6	376
24	D	50	32½	1⅛	29¾	16	1	74·3	225
24	E	100	32½	1½	29¾	16	1	82·4	300
24	F	150	33½	1⅝	30¾	24	1¼	106·7	375
24	H	250	33½	2¼	30¾	24	1¼	154·8	517
24	J	350	33½	2¾	30¾	24	1⅜	210	631

TABLE XIX (Sect. 162). HEAT CONDUCTIVITY

MATERIAL		APPROXIMATE BTU/SQ. FT./HOUR/ °F. DIFF./1" THICKNESS
Heating Surfaces	Copper	2,620
	Aluminium	1,430
	Brass	720
	Cast iron	340
	Steel	310
	Lead	240
Resistant Films	Water at 32° F.	4
	Water at 200° F.	5
	Air	·2
	Scale	1 to 12
Lagging Materials	Diatomite, Kieselguhr or fossil meal — Poor	1·0
	Diatomite, Kieselguhr or fossil meal — Good	·6
	85% magnesia	·4
	Asbestos flock	·4
	Glass wool	·4
	Animal hair	·3
	Cork	·3

TABLE XX (Sect. 165). APPROXIMATE OUTSIDE TEMPERATURES OF LAGGED SURFACES 6 IN. BORE STEEL PIPE, STILL AIR AT 70° F.

INTERNAL TEMPERATURE °F.	LAGGING THICKNESS					
	1"	1½"	2"	2½"	3"	4"
100	75	74	73	72	72	71
150	83	80	78	76	75	74
200	91	86	83	80	79	76
250	99	92	88	84	82	79
300	107	98	93	88	85	82
350	115	105	98	92	88	84
400	123	111	102	96	91	87
450	131	117	107	100	95	89
500	139	123	112	104	98	92
600	155	135	122	112	104	98
700	171	147	132	120	110	102
800	188	160	142	128	117	107

TABLE XXI (Sect. 165). RECOMMENDED LAGGING THICKNESSES

INTERNAL TEMPERATURE °F.	THICKNESS OF LAGGING				
	HIGH TEMPERATURE COMPOUND	85% MAGNESIA OR ASBESTOS			
		PIPE BELOW 3″ DIA.	PIPE 3″ TO 6″ DIA.	PIPE 6″ TO 9″ DIA.	PIPE OVER 9″ DIA. AND FLAT
Up to 200	—	1″	1″	1″	1″
200–300	—	1″	1″	1½″	2″
300–400	—	1″	1½″	2″	2½″
400–500	—	1½″	2″	2″	2½″
500–600	—	1½″	2″	2½″	3″
600–700	½″ plus	2″	2″	2½″	3″
700–800	1″ plus	2″	2″	2½″	3″

TABLE XXII (Sect. 166). APPROXIMATE HEAT LOSSES FROM BARE AND LAGGED FLAT SURFACES IN STILL AIR AT 70°F.

INTERNAL TEMPERATURE °F.	BTU/SQ. FT./HOUR						
	BARE	LAGGING THICKNESS					
		1″	1½″	2″	2½″	3″	4″
100	60	10	7	—	—	—	—
150	155	25	20	—	—	—	—
200	295	45	30	25	—	—	—
250	450	65	45	35	—	—	—
300	645	85	60	45	40	—	—
350	875	105	75	60	45	—	—
400	1,140	125	90	70	55	45	—
450	1,450	150	105	80	65	55	—
500	1,810	175	125	95	75	65	—
600	2,670	—	160	120	100	85	65
700	3,750	—	—	150	125	105	80
800	5,100	—	—	185	150	130	100

TABLE XXIII (Sect. 166). APPROXIMATE HEAT LOSSES FROM BARE AND LAGGED 6-IN. PIPE IN STILL AIR AT 70° F.

INTERNAL TEMPERATURE °F.	BTU/SQ. FT./HOUR						
	LAGGING THICKNESS						
	BARE	1″	1½″	2″	2½″	3″	4″
100	58	11	8·5	—	—	—	—
150	166	31	23	—	—	—	—
200	296	52	39	32	—	—	—
250	454	74	55	45	—	—	—
300	646	98	73	60	51	—	—
350	876	122	91	75	64	—	—
400	1,142	148	110	90	77	68	—
450	1,452	176	131	106	91	80	—
500	1,810	204	152	123	105	93	—
600	2,671	—	221	160	136	120	100
700	3,750	—	—	200	170	150	124
800	5,110	—	—	242	207	182	151

TABLE XXIV (Sect. 166). AREAS OF PIPE SURFACES

BORE	SQ. FT. (STEEL PIPE UP TO 250 PSI)	
	PER FOOT LENGTH PARALLEL PIPE	FLANGE (2 EDGES 2 SIDES)
½″	·221	·331
¾″	·278	·327
1″	·352	·386
1½″	·500	·488
2″	·622	·655
2½″	·785	·760
3″	·916	·931
4″	1·178	1·134
5″	1·440	1·638
6″	1·710	1·810
7″	1·963	2·169
8″	2·23	2·45
9″	2·49	2·98
10″	2·75	3·17
12″	3·27	3·85
15″	4·19	4·94
18″	4·98	6·22
24″	6·55	9·38

TABLE XXV (Sect. 168). LENGTH OF ASBESTOS ROPE NEEDED TO WRAP PIPES

PIPE BORE	LENGTH IN FEET PER FOOT OF PIPE				
	½" ROPE	¾" ROPE	1" ROPE	1½" ROPE	2" ROPE
½"	11				
¾"	12	10			
1"	14	11	10		
1½"	17	14	12	10	
2"	20	16	14	11	10
3"	27	21	17	14	12
4"	34	25	20	16	14
6"	47	34	27	20	17

TABLE XXVI (Sect. 170). APPROXIMATE EFFECT OF AIR MOVEMENT ON HEAT LOSS

RELATIVE HEAT LOSS	AIR VELOCITY	
	FEET PER SECOND	MILES PER HOUR
1	Still	
1·5	4	3
2	8·5	6
2·5	13	9
3	18	12
3·5	24	16
4	30	21

TABLE XXVII (Sect. 172). RECOMMENDED VELOCITIES IN PIPES

High vacuum vapour	200 – 350 feet per second
Moderate vacuum vapour	150 – 200 „ „ „
Exhaust steam—wet	70 – 100 „ „ „
Dry saturated steam	100 – 130 „ „ „
Superheated steam	150 – 200 „ „ „
Water	4 – 8 „ „ „

TABLE XXIII (Sect. 166). APPROXIMATE HEAT LOSSES FROM BARE AND LAGGED 6-IN. PIPE IN STILL AIR AT 70° F.

INTERNAL TEMPERATURE °F.	BTU/SQ. FT./HOUR						
		LAGGING THICKNESS					
	BARE	1″	1½″	2″	2½″	3″	4″
100	58	11	8·5	—	—	—	—
150	166	31	23	—	—	—	—
200	296	52	39	32	—	—	—
250	454	74	55	45	—	—	—
300	646	98	73	60	51	—	—
350	876	122	91	75	64	—	—
400	1,142	148	110	90	77	68	—
450	1,452	176	131	106	91	80	—
500	1,810	204	152	123	105	93	—
600	2,671	—	221	160	136	120	100
700	3,750	—	—	200	170	150	124
800	5,110	—	—	242	207	182	151

TABLE XXIV (Sect. 166). AREAS OF PIPE SURFACES

BORE	SQ. FT. (STEEL PIPE UP TO 250 PSI)	
	PER FOOT LENGTH PARALLEL PIPE	FLANGE (2 EDGES 2 SIDES)
½″	·221	·331
¾″	·278	·327
1″	·352	·386
1½″	·500	·488
2″	·622	·655
2½″	·785	·760
3″	·916	·931
4″	1·178	1·134
5″	1·440	1·638
6″	1·710	1·810
7″	1·963	2·169
8″	2·23	2·45
9″	2·49	2·98
10″	2·75	3·17
12″	3·27	3·85
15″	4·19	4·94
18″	4·98	6·22
24″	6·55	9·38

TABLE XXV (Sect. 168). LENGTH OF ASBESTOS ROPE NEEDED TO WRAP PIPES

PIPE BORE	LENGTH IN FEET PER FOOT OF PIPE				
	½" ROPE	¾" ROPE	1" ROPE	1¼" ROPE	2" ROPE
½"	11				
¾"	12	10			
1"	14	11	10		
1½"	17	14	12	10	
2"	20	16	14	11	10
3"	27	21	17	14	12
4"	34	25	20	16	14
6"	47	34	27	20	17

TABLE XXVI (Sect. 170). APPROXIMATE EFFECT OF AIR MOVEMENT ON HEAT LOSS

RELATIVE HEAT LOSS	AIR VELOCITY	
	FEET PER SECOND	MILES PER HOUR
1	Still	
1·5	4	3
2	8·5	6
2·5	13	9
3	18	12
3·5	24	16
4	30	21

TABLE XXVII (Sect. 172). RECOMMENDED VELOCITIES IN PIPES

High vacuum vapour	200 – 350 feet per second
Moderate vacuum vapour	150 – 200 „ „ „
Exhaust steam—wet	70 – 100 „ „ „
Dry saturated steam	100 – 130 „ „ „
Superheated steam	150 – 200 „ „ „
Water	4 – 8 „ „ „

TABLE XXVIII (Sect. 173). APPROXIMATE RESISTANCE OF PIPE FITTINGS

PIPE BORE INCHES	EQUIVALENT STRAIGHT LENGTH IN FEET							
	STANDARD ELBOW	STANDARD BEND	TEE		OPEN VALVE			EXPANSION LOOP
			BARREL	BRANCH	SLIDE	ANGLE	GLOBE	
1	2	·5	·5	2·25	·5	1·5	3·25	2·25
1¼	3	1	1	3·5	·75	2·5	5·5	3·5
2	4	1·25	1·25	5	1	3·5	7·5	5
2½	5	1·5	1·5	6·5	1·25	4·5	10	6·5
3	6	2	2	8·5	1·5	5·5	12	8·25
4	8	3	3	12	2·25	8	18	12
5	10	4	4	15	3	10	23	15
6	13	5	5	19	3·5	13	29	19
7	15	6	6	23	4·25	15	34	23
8	17	7	7	27	5	18	40	27
9	19	8	8	31	6	21	46	31
10	21	9	9	35	7	24	53	35
12	25	11	11	44	8	30	66	44
15	31	14	14	57	11	34	86	57

TABLE XXIX (Sect. 174). STEAM VELOCITIES IN PIPES

PIPE SIZE	PRESSURE	VELOCITY—FEET PER SECOND WITH PRESSURE DROP/100 FT. OF :—						
		·25 PSI	·5 PSI	·75 PSI	1·0 PSI	1·5 PSI	2·0 PSI	3·0 PSI
3″ pipe	20″ vac.	135	180	235	270	330	395	500
,, ,,	Atmos.	85	120	145	170	205	250	320
,, ,,	25 psi.g.	50	75	85	105	130	150	190
,, ,,	100 ,,	30	45	55	65	75	90	115
,, ,,	400 ,,			30	35	40	50	60
6″ pipe	20″ vac.	220	315	400	460	550	625	750
,, ,,	Atmos.	145	190	240	300	350	405	500
,, ,,	25 psi.g.	80	115	145	170	200	240	300
,, ,,	100 ,,	50	70	90	105	130	150	180
,, ,,	400 ,,		30	45	55	65	75	95

TABLE XXX (Sect. 192). THERMAL EXPANSION OF PIPES

TEMPERATURE RISE °F.	EXPANSION IN INCHES PER 100 FEET
From 60 to 150	·75
60 200	1·15
60 250	1·6
60 300	2·0
60 350	2·4
60 400	2·9
60 450	3·3
60 500	3·8
60 600	4·8
60 700	5·8
60 800	6·9
60 900	8·0

TABLE XXXI (Sect. 204). APPROXIMATE COST OF BUYING, ERECTING, LAGGING AND SHEETING 100 FT. OF STRAIGHT STEEL PIPING WITH WELDED JOINTS (1957)

NOMINAL PIPE DIAMETER	COST OF 100 FT. OF ERECTED PIPE		LAGGING $1\frac{1}{2}''$ THICK AND SHEETING		TOTAL
	PIPE	LABOUR	MATERIAL	LABOUR	
INCHES	£ s. d.	£ s. d.	£ s. d.	£ s. d.	£ s. d.
2	11 18 5	8 17 9	28 7 0	10 2 0	59 5 2
3	17 8 5	13 6 6	33 18 2	12 12 6	77 15 7
4	31 3 10	16 15 9	39 2 8	15 3 0	102 5 3
5	37 0 2	22 14 9	44 1 8	19 4 3	123 0 10
6	52 19 4	25 13 6	49 4 5	23 5 6	151 2 9
7	68 15 3	33 11 6	54 5 0	28 6 6	184 18 3
8	89 12 0	39 10 0	59 14 4	32 17 9	221 14 1
9	97 10 4	45 18 5	64 16 2	36 9 0	244 13 11
10	112 2 1	51 7 0	69 16 0	40 10 3	273 15 4

TABLE XXXII (Sect. 206). PRESSURE EQUIVALENTS

PSI	INCH HG.	FOOT H_2O
·4332	·8819	1
·4912	1	1·134
·8664	1·7638	2
·9824	2	2·268
1	2·036	2·309
1·2996	2·646	3
1·4736	3	3·402
1·7328	3·528	4
1·9648	4	4·536
2	4·072	4·618
2·1660	4·4095	5
2·4560	5	5·670
2·5992	5·2914	6
2·9472	6	6·804
3	6·108	6·927
3·0324	6·1733	7
3·4384	7	7·938
3·4656	7·0552	8
3·8988	7·9371	9
3·9296	8	9·072
4	8·144	9·236
4·4208	9	10·206
5	10·180	11·545
6	12·216	13·854
7	14·252	16·163
8	16·288	18·472
9	18·324	20·781

TABLE XXXIII (Sect. 229). SCALE FOR MERCURY U-TUBE METER

TABLE XXXIV (Sect. 230). FACTOR K FOR ORIFICES

TABLE XXXV (Sect. 230). PIPE SIZES FOR METERING STEAM BY ORIFICE

TABLE XXXVI (Sect. 232). UNRECOVERED STEAM PRESSURE DROP ACROSS ORIFICE

ORIFICE RATIO $\frac{d}{D}$	LOSS OF PRESSURE DROP	
	%	12″ MERCURY METER PSI
·2	95	5·6
·25	92	5·4
·3	89	5·2
·35	86	5·1
·4	83	4·9
·45	79	4·6
·5	74	4·4
·55	69	4·1
·6	64	3·8
·65	58	3·4
·7	52	3·1

TABLE XXXVII (Sect. 236). FLOW OF WATER IN LB. PER HOUR THROUGH AN ORIFICE

HEAD *h*. INCHES	ORIFICE DIAMETER *d*.													
	¼″	⅜″	½″	⅝″	¾″	⅞″	1″	1¼″	1½″	1¾″	2″	2½″	3″	4″
3	183	411	730	1,141	1,643	2,236	2,920	4,563	6,570	8,943	11,681	18,251		
4	211	474	843	1,317	1,897	2,582	3,372	5,269	7,587	10,327	13,488	21,075	30,348	
5	236	530	942	1,473	2,121	2,886	3,770	5,890	8,482	11,545	15,080	23,562	33,929	60,318
6	258	581	1,032	1,613	2,323	3,161	4,129	6,452	9,290	12,645	16,518	25,806	37,161	66,064
7	279	627	1,115	1,743	2,509	3,416	4,461	6,971	10,038	13,662	17,845	27,882	40,150	71,378
8	298	671	1,192	1,863	2,682	3,651	4,768	7,450	10,728	14,602	19,072	29,800	42,912	76,288
9	316	711	1,265	1,976	2,845	3,873	5,058	7,903	11,380	15,490	20,232	31,613	45,522	80,928
10	333	750	1,333	2,082	2,999	4,082	5,331	8,330	11,995	16,327	21,325	33,320	47,980	85,298
11	350	786	1,398	2,185	3,146	4,282	5,592	8,738	12,583	17,127	22,370	34,953	50,332	89,479
12	365	821	1,460	2,281	3,285	4,472	5,840	9,125	13,141	17,886	23,361	36,502	52,563	93,445
13	380	855	1,520	2,375	3,420	4,655	6,080	9,500	13,679	18,619	24,319	37,998	54,717	97,275
14	394	887	1,577	2,464	3,549	4,830	6,309	9,859	14,195	19,321	25,236	39,431	56,781	100,944
15	408	918	1,632	2,551	3,673	4,999	6,530	10,203	14,692	20,000	26,120	40,812	58,769	104,478
16	422	948	1,686	2,634	3,794	5,163	6,744	10,538	15,174	20,654	26,976	42,150	60,696	107,904
17	435	978	1,738	2,715	3,910	5,322	6,951	10,862	15,641	21,289	27,806	43,446	62,562	111,222
18	447	1,006	1,788	2,794	4,024	5,477	7,154	11,178	16,096	21,908	28,615	44,711	64,383	114,459
19	459	1,033	1,837	2,871	4,134	5,627	7,349	11,483	16,536	22,507	29,397	45,933	66,143	117,588
20	471	1,060	1,885	2,945	4,241	5,773	7,540	11,781	16,965	23,091	30,159	47,124	67,858	120,637
21	483	1,087	1,932	3,018	4,346	5,916	7,727	12,073	17,386	23,664	30,908	48,293	69,542	123,631
22	494	1,112	1,977	3,089	4,448	6,054	7,907	12,355	17,792	24,216	31,629	49,421	71,166	126,517
23	505	1,137	2,022	3,159	4,548	6,191	8,086	12,634	18,194	24,764	32,344	50,538	72,775	129,377
24	516	1,162	2,065	3,226	4,646	6,324	8,260	12,906	18,584	25,295	33,039	51,623	74,337	132,155
25	527	1,185	2,108	3,293	4,742	6,454	8,430	13,172	18,968	25,817	33,720	52,688	75,870	134,880

TABLE XXXVIII (Sect. 237). WATER FLOW THROUGH V-NOTCH METER IN GALLONS PER HOUR

LIQUID CREST HEIGHT INCHES "h"	NOTCH ANGLE "θ"					
	10°	20°	30°	45°	60°	90°
1	10	21	32	49	68	117
2	58	116	176	272	378	654
4	324	649	984	1,517	2,110	3,645
8	1,806	3,622	5,490	8,460	11,770	20,340
16	10,070	20,210	30,620	47,210	65,680	113,500

TABLE XXXIX (Sect. 240). APPROXIMATE OVERALL THERMAL EFFICIENCIES OF ENGINES AND TURBINES

(Coal to Useful Power Produced)

TYPE OF MACHINE	EFFICIENCY
	Per cent.
Back pressure turbine, efficient, 5,000 kW	73
Back pressure engine, bad, 100 kW	30
Condensing turbine, 100,000 kW	31
Average British grid (1957)	25
Condensing turbine, medium pressure, 5,000 kW ..	15
Condensing engine, industrial, 1,000 H.P.	9
Colliery winding engine, good	8
Express locomotive	7
Shunting locomotive	4
Colliery winding engine, bad	3
Non-expansive non-condensing pump	2

TABLE XL (Sect. 241). ENGINE STEAM CONSUMPTION DUE TO PUMPING AGAINST BACK PRESSURE

TABLE XLI (Sect. 246). EFFECT OF SUPERHEAT ON STEAM CONSUMPTION OF SMALL RECIPROCATING ENGINE

(Ripper)

APPROX. SUPERHEAT °F.	STEAM CONSUMPTION LB./H.P./HOUR	RELATIVE EFFICIENCY IMPROVEMENT PER CENT	ADIABATIC HEAT DROP BTU	RELATIVE RANKINE IMPROVEMENT PER CENT
Sat.	39·6	—	140	—
100	33·8	15	151	8
150	29	25	159	14
200	25	36	166	19
250	23·4	44	171	22
300	20·1	49	187	33

TABLE XLII (Sect. 253). APPROXIMATE EFFICIENCY RATIOS OF SMALL ENGINES

TYPE OF ENGINE	WITHOUT SUPERHEAT	WITH SUPERHEAT
Simple non-condensing up to 50 I.H.P.	43	50
Simple non-condensing up to 300 I.H.P.	60	70
Simple condensing up to 50 I.H.P.	41	48
Simple condensing up to 300 I.H.P.	55	65
Compound condensing up to 50 I.H.P.	55	65
Compound condensing up to 300 I.H.P.	70	75

The figures given in Table XLII, in so far as they can be compared, are rather more optimistic than those in Table X.

TABLE XLIII (Sect. 294). CHARACTERISTICS OF STEAM TRAPS

TYPE OF TRAP	TYPE OF DISCHARGE	OPENING FORCE	CLOSING FORCE	TEMPERATURE OF CONDENSATE	DISCHARGE AIR	WITHSTAND WATER HAMMER	STRAINER BEFORE TRAP	CONDENSATE DRAINED	WILL LIFT CONDENSATE	DAMAGE BY FROST	CHECK VALVE BEFORE TRAP	SUITABLE FOR SUPERHEATED STEAM	SUITABLE FOR VARYING PRESSURE
Plain float.. ..	Continuous	Buoyancy	Float weight	Saturation	No	No	Highly desirable	Instantly	Yes	Yes	No	Yes	Yes
Trip float	Intermittent	Buoyancy	Float weight	Saturation	No	No	Desirable	As formed	Yes	Yes	No	Yes	Yes
Open bucket ..	Intermittent	Weight of bucket	Buoyancy	Saturation	No	Yes	Not essential	As formed	Yes	Yes	Yes	Yes†	Yes
Inverted bucket ..	Intermittent	Weight of bucket	Buoyancy	Saturation	Yes	Yes	Not essential	As formed	Yes	Yes	Yes	Yes†	Yes
Metallic expansion	Semi-continuous	Metallic contraction	Metallic expansion	Pre-set temperature	Yes	Yes	Highly desirable	At pre-set temperature	Yes	No	No	Yes	No
Liquid expansion..	Semi-continuous	Steam pressure	Liquid expansion	Pre-set temperature	Yes	Yes	Highly desirable	At pre-set temperature	Yes	No	No	Yes	No
Balanced pressure expansion.	Semi-continuous	Differential pressure	Differential pressure	Below saturation	Yes	No	Highly desirable	After cooling	Yes	No	No	No	Yes
Relay—float, bucket, bottle.	Continuous if compensated	Outside source unlimited	Outside source unlimited	Saturation	No	—	Desirable	As formed	Yes	Yes	No	Yes	Yes
Pumping or lifting	Intermittent	—	—	Any temperature below steam temperature	No	—	Not essential	As formed	Yes	Yes	In trap	Yes	Yes
Barometric leg* ..	Continuous	Always open		212°F. or less	No	—	Unnecessary	Instantly	No	—	No	—	Yes
U-tube*	Continuous	Always open		Saturation at exit pressure	No	—	Unnecessary	Instantly	No	—	No	—	No

* These are not strictly traps; they are piping systems which replace traps—see Chapter 9.
† The water sealing the trap may, in exceptional cases, be evaporated—see Section 284.

TABLE XLIV (Sect. 312). SUCTION HEAD FOR PUMPING HOT WATER

(Weir)

TEMPERATURE OF WATER °F.	SUCTION LIFT FEET	PRESSURE HEAD FEET
130	10	
150	7	
170	2	
175	Level	Level
190		5
200		10
210		15
212		17

TABLE XLV (Sect. 329). HEAT TRANSFER WITH VARIOUS RESISTANT FILMS

TABLE XLVI (Sect. 331). PUBLISHED OVERALL HEAT TRANSFER RATES

PLANT	CONDITIONS	BTU/SQ. FT./HOUR/°F. DIFF.
Condenser	Water flowing at 1 ft./sec.	150 to 400
Condenser	Water flowing at 2 ft./sec.	200 to 550
Condenser	Water flowing at 4 ft./sec.	250 to 750
Boiler	Gas to water	2 to 8
Economiser	Gas to water	1 to 5
Superheaters	Gas to steam	2 to 6
Tank coils	Temp. difference 50° F.	100 to 225
Tank coils	Temp. difference 100° F.	175 to 300
Tank coils	Temp. difference 200° F.	225 to 475
Air heater	Convection only	2
Air heater	Air velocity 1 ft./sec. Temp. diff. 100° F.	3
Air heater	Air velocity 5 ft./sec. Temp. diff. 100° F.	5
Air heater	Air velocity 10 ft./sec. Temp. diff. 100° F.	8

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TABLE XLVII (Sect. 331). REASONABLE PRACTICAL HEAT TRANSFER RATES

OPERATION (ON WATER OR DILUTE LIQUORS)	BTU/SQ. FT./HOUR/°F. DIFF.
Calorifiers and condensers with low velocity liquid	150
Calorifiers and condensers with high velocity liquid	300
Tank coils, low pressure with natural circulation	100
Tank coils, high pressure and natural circulation	200
Tank coils, low pressure with assisted circulation	200
Tank coils, high pressure with assisted circulation	300
Natural circulation evaporators with low pressure steam ..	300
Natural circulation evaporators with high pressure steam ..	500
Assisted circulation evaporators	750
Space heating by water—convection only	1·5
Space heating by steam—convection only	2
Space heating by steam—10 ft./sec. air velocity	8

TABLE XLVIII (Sect. 334). EFFECT OF WATER VELOCITY ON HEAT TRANSFER RATE

(Turbulent Flow)

RELATIVE LIQUID VELOCITY	HEAT TRANSFER RATE
1	100
2	174
3	241
4	303
5	362
6	419
7	474
8	527
9	580
10	630

TABLE XLIX (Sect. 345). FEET LENGTH PER SQUARE FOOT OF SURFACE OF TUBES OF SMALL DIAMETER

DIAMETER	·00″	·01″	·02″	·03″	·04″	·05″	·06″	·07″	·08″	·09″
0″		382	191	127	95·5	76·4	63·7	54·6	47·7	42·4
·1″	38·2	34·7	31·8	29·4	27·3	25·5	23·9	22·5	21·2	20·1
·2″	19·10	18·19	17·36	16·61	15·92	15·28	14·69	14·15	13·64	13·17
·3″	12·73	12·32	11·94	11·57	11·23	10·91	10·61	10·32	10·05	9·79
·4″	9·55	9·32	9·09	8·88	8·68	9·49	8·30	8·13	7·96	7·80
·5″	7·64	7·49	7·35	7·21	7·07	6·94	6·82	6·70	6·59	6·47
·6″	6·37	6·26	6·16	6·06	5·97	5·88	5·79	5·70	5·62	5·54
·7″	5·46	5·38	5·31	5·23	5·16	5·09	5·02	4·96	4·90	4·84
·8″	4·77	4·72	4·66	4·60	4·55	4·49	4·44	4·39	4·34	4·29
·9″	4·24	4·20	4·15	4·11	4·06	4·02	3·98	3·94	3·90	3·86
1·0″	3·82	3·78	3·74	3·71	3·67	3·64	3·60	3·57	3·54	3·50
1·1″	3·47	3·44	3·41	3·38	3·35	3·32	3·29	3·26	3·24	3·21
1·2″	3·18	3·16	3·13	3·11	3·08	3·06	3·03	3·01	2·98	2·96
1·3″	2·94	2·92	2·89	2·87	2·85	2·83	2·82	2·80	2·78	2·76
1·4″	2·73	2·71	2·69	2·67	2·65	2·63	2·62	2·60	2·58	2·56
1·5″	2·55	2·53	2·51	2·50	2·48	2·46	2·45	2·43	2·42	2·40
1·6″	2·39	2·37	2·36	2·34	2·33	2·31	2·30	2·29	2·27	2·26
1·7″	2·25	2·23	2·22	2·21	2·20	2·18	2·17	2·16	2·15	2·13
1·8″	2·12	2·11	2·10	2·09	2·08	2·06	2·05	2·04	2·03	2·02
1·9″	2·01	2·00	1·99	1·98	1·97	1·96	1·95	1·94	1·93	1·92
2·0″	1·91	1·90	1·89	1·88	1·87	1·86	1·85	1·85	1·84	1·83
2·1″	1·82	1·81	1·80	1·79	1·78	1·78	1·77	1·76	1·75	1·74
2·2″	1·74	1·73	1·72	1·71	1·71	1·70	1·69	1·68	1·68	1·67
2·3″	1·66	1·65	1·64	1·64	1·63	1·63	1·62	1·61	1·60	1·60
2·4″	1·59	1·58	1·58	1·57	1·57	1·56	1·55	1·55	1·54	1·53
2·5″	1·53	1·52	1·52	1·51	1·50	1·50	1·49	1·49	1·48	1·47
2·6″	1·47	1·46	1·46	1·45	1·45	1·44	1·44	1·43	1·43	1·42
2·7″	1·41	1·41	1·40	1·40	1·39	1·39	1·38	1·38	1·37	1·37
2·8″	1·36	1·36	1·35	1·35	1·34	1·34	1·34	1·33	1·33	1·32
2·9″	1·32	1·31	1·31	1·30	1·30	1·29	1·29	1·29	1·28	1·28

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TABLE L (Sect. 345). SQUARE FEET OF SURFACE PER FOOT OF LENGTH OF TUBES OF LARGE DIAMETER

DIAMETER	·0″	·1″	·2″	·3″	·4″	·5″	·6″	·7″	·8″	·9″
3·0″	·787	·811	·838	·864	·890	·916	·942	·969	·995	1·021
4·0″	1·047	1·073	1·100	1·126	1·152	1·178	1·204	1·230	1·257	1·283
5·0″	1·309	1·335	1·361	1·388	1·414	1·440	1·466	1·492	1·518	1·545
6·0″	1·570	1·597	1·623	1·649	1·676	1·702	1·728	1·754	1·780	1·806
7·0″	1·833	1·859	1·885	1·911	1·937	1·963	1·990	2·016	2·042	2·068
8·0″	2·09	2·12	2·15	2·17	2·20	2·23	2·25	2·28	2·30	2·33
9·0″	2·36	2·38	2·41	2·43	2·46	2·49	2·51	2·54	2·57	2·59
10·0″	2·62	2·64	2·67	2·70	2·72	2·75	2·78	2·80	2·83	2·85
11·0″	2·88	2·91	2·93	2·96	2·98	3·01	3·04	3·06	3·09	3·12
12·0″	3·14	3·17	3·19	3·22	3·25	3·27	3·30	3·32	3·35	3·38
13·0″	3·40	3·43	3·46	3·48	3·51	3·53	3·56	3·59	3·61	3·64
14·0″	3·67	3·69	3·72	3·74	3·77	3·80	3·82	3·85	3·87	3·90
15·0″	3·93	3·95	3·98	4·01	4·03	4·06	4·08	4·11	4·14	4·16
16·0″	4·19	4·21	4·24	4·27	4·29	4·32	4·35	4·37	4·40	4·42
17·0″	4·45	4·48	4·50	4·53	4·56	4·58	4·61	4·63	4·66	4·69
18·0″	4·71	4·74	4·76	4·79	4·82	4·84	4·87	4·90	4·92	4·95
19·0″	4·97	5·00	5·03	5·05	5·08	5·11	5·13	5·16	5·18	5·21
20·0″	5·24	5·26	5·29	5·31	5·34	5·37	5·39	5·42	5·45	5·47
21·0″	5·50	5·52	5·55	5·58	5·60	5·63	5·65	5·68	5·71	5·73
22·0″	5·76	5·79	5·81	5·84	5·86	5·89	5·92	5·94	5·97	6·00
23·0″	6·02	6·05	6·07	6·10	6·13	6·15	6·18	6·20	6·23	6·26
24·0″	6·28	6·31	6·34	6·36	6·39	6·41	6·44	6·47	6·49	6·52

TABLE LI (Sect. 352). SURFACE/VOLUME RATIOS OF SOME PANS AND EVAPORATORS

TABLE LII (Sect. 355). VACUA AND BOILING POINTS OF 60 PER CENT. SUGAR SOLUTIONS

TABLE LIII (Sect. 371). HEAT TRANSFER RATE IN QUADRUPLE EFFECT EVAPORATORS ON SUGAR CANE JUICE

(Kerr)

	HEAT TRANSFER RATE BTU/SQ. FT./HOUR/°F. DIFF.				
	1ST EFFECT	2ND EFFECT	3RD EFFECT	4TH EFFECT	OVERALL
Standard	205	280	240	40	191
Horizontal	290	237	130	76	183
Lillie	427	427	395	158	351
Kestner	310	325	295	155	277

TABLE LIV (Sect. 392). HIGH v. LOW PRESSURE HEATING STEAM—CONSTANT QUANTITY

TABLE LV (Sect. 392). HIGH v. LOW PRESSURE HEATING STEAM—CONSTANT HEATING

TABLE LVI (Sect. 393). PERMISSIBLE WEIGHT OF FLASH STEAM FROM WATER SURFACE
(Ruths)

PRESSURE ON WATER SURFACE	LB. FLASH STEAM/HOUR/SQ. FT. WATER SURFACE
Atmos.	44
5 psi.g.	59
10 „	74
20 „	104
30 „	134
40 „	164
50 „	194
100 „	344
150 „	494
200 „	644

TABLE LVII (Sect. 393). STEAM VELOCITIES EXERTING FORCES ON WATER DROPS EQUAL TO THEIR WEIGHTS
(Hausbrand)

STEAM PRESSURE	STEAM VELOCITY IN FT./SEC. ON DROPS OF DIAMETER :—							
	·005-IN.	·01-IN.	·02-IN.	·05-IN.	·1-IN.	·15-IN.	·2-IN.	·25-IN.
25-in. vac.	13·1	22·6	32·0	50·6	72·1	88·2	101·3	110·5
20-in. „	9·6	16·6	23·5	37·2	52·9	64·8	74·4	81·2
15-in. „	7·9	13·6	19·3	30·5	43·7	53·2	61·1	66·6
10-in. „	6·9	11·9	16·9	26·8	38·1	46·6	53·5	58·4
5-in. „	6·2	10·7	15·2	24·0	34·2	41·9	48·1	52·5
Atmos.	5·7	9·8	13·9	22·0	31·3	38·3	44·0	48·0
10 psi.g.	3·5	6·0	8·6	13·5	19·2	23·5	27·0	29·5
20 „	2·6	4·4	6·2	9·8	14·0	17·1	19·6	21·4
30 „	2·0	3·5	4·9	7·8	11·1	13·6	15·6	17·0
40 „	1·7	2·9	4·1	6·4	9·1	11·1	12·8	14·0
50 „	1·4	2·4	3·5	5·5	7·8	9·6	11·0	12·0

TABLE LVIII (Sect. 404). TEMPERATURE OF STEAM ADULTERATED WITH AIR

STEAM PRESSURE	TEMPERATURES °F OF SATURATED STEAM MIXED WITH AIR OF PER CENT. BY VOLUME										
	0%	5%	10%	20%	30%	40%	50%	60%	70%	80%	90%
25-in. vac.	134	132	130	125	120	115	109	101	92	79	59
20-in. ,,	161	159	157	152	147	141	134	125	115	101	79
15-in. ,,	179	177	174	169	163	157	150	141	130	115	92
10-in. ,,	192	190	187	182	176	169	161	152	141	125	101
5-in. ,,	203	201	198	192	186	179	171	161	150	134	109
Atmos.	212	209	207	201	195	187	179	169	157	141	115
10 psi.g.	239	237	234	227	220	212	203	193	179	162	134
20 ,,	259	256	253	246	239	230	221	209	195	177	147
30 ,,	274	271	268	261	253	244	234	222	207	188	158
40 ,,	287	283	280	273	264	255	245	233	218	198	166
50 ,,	298	294	291	283	275	265	255	242	226	206	173
60 ,,	307	304	300	293	284	274	263	250	234	213	180
70 ,,	316	313	309	300	292	282	271	257	241	219	186
80 ,,	324	320	316	308	299	289	278	264	247	225	191
90 ,,	331	328	324	315	306	296	284	270	253	231	195
100 ,,	338	334	330	322	312	302	290	276	258	235	200

TABLE LIX (Sect. 414). POUNDS OF JET CONDENSER WATER NEEDED FOR DIFFERENT VACUA AND FOR DIFFERENT AIR PERCENTAGES AND TEMPERATURE DIFFERENCES

	LB. WATER AT 60° F. PER LB. VAPOUR											
PER CENT. AIR BY VOLUME	0 PER CENT.			10 PER CENT.			20 PER CENT.			30 PER CENT.		
TEMP. DIFF.	0° F.	5° F.	10° F.	0° F.	5° F.	10° F.	0° F.	5° F.	10° F.	0° F.	5° F.	10° F.
VACUUM												
Atmos.	6 4	6·6	6·8	6·6	7·0	7·2	6·9	7·2	7·5	7·3	7·6	7·9
5-in.	6·8	7·1	7·4	7·1	7·4	7·7	7·5	7·8	8·1	8·2	8·6	9·0
10-in.	7·5	7·8	8·1	7·8	8·1	8·5	8·1	8·5	8·9	8·6	9·0	9·5
15-in.	8·3	8·7	9·2	8·7	9·2	9·7	9·1	9·6	10·2	9·7	10·3	10·9
20-in.	9·9	10·5	11·1	10·4	11·0	11·7	11·0	11·6	12·4	11·8	12·5	13·4
21-in.	10·6	11·2	11·9	10·8	11·5	12·3	11·5	12·2	13·1	12·4	13·2	14·2
22-in.	11·0	11·6	12·4	11·5	12·2	13·1	12·2	13·1	14·0	13·0	14·0	15·1
23-in.	11·6	12·4	13·3	12·4	13·2	14·2	13·0	14·0	15·1	14·0	15·1	16·3
24-in.	12·5	13·4	14·3	13·2	14·2	15·3	14·2	15·3	16·6	15·3	16·6	18·1
25-in.	13·8	14·8	16·1	14·6	15·8	17·2	15·7	17·1	18·8	17·1	18·8	20·7
26-in.	15·7	17·1	18·8	16·8	18·4	20·3	18 0	19·9	22·1	19·4	21·6	24·2
27-in.	18·7	20·7	23·1	20·2	22·5	25·4	22·0	24·7	28·2	24·1	27·4	31·7
28-in.	25·3	28·9	33·8	27·3	31·6	37·4	30·6	36·0	43·8	36·0	43·7	55·4
28·5-in.	32·6	38·8	47·8	37·3	45·6	58·6	41·8	52·5	70·3	52·4	70·2	105·8
29-in.	55·2	75·3	117·6	65·7	96·0	176·8	87·8	151·1	531·5	131·9	353·2	—

TABLE LX (Sect. 443). CAPACITY OF CYLINDRICAL TANKS OF VARIOUS DIAMETERS

Gallons per Foot of Length

DIAMETER	0	1-in.	2-in.	3-in.	4-in.	5-in.	6-in.	7-in.	8-in.	9-in.	10-in.	11-in.
FEET												
0	—	·0340	·1359	·3058	·5436	·8493	1·223	1·665	2·174	2·752	3·397	4·111
1	4·892	5·741	6·659	7·644	8·697	9·818	10·07	12·26	13·59	14·98	16·44	17·97
2	19·57	21·23	22·97	24·77	26·63	28·57	30·58	32·65	34·79	37·00	39·27	41·62
3	44·03	46·51	49·06	51·67	54·35	57·11	59·93	62·82	65·77	68·79	71·89	75·04
4	78·28	81·57	84·93	88·36	91·86	95·43	99·06	102·8	106·5	110·4	114·3	118·3
5	122·3	126·4	130·6	134·8	139·2	143·5	148·0	152·5	157·1	161·7	166·5	171·3
6	176·1	181·0	186·0	191·1	196·2	201·4	206·7	212·[illegible]	217·4	222·9	228·4	234·0
7	239·7	245·5	251·3	257·1	263·1	269·1	275·2	281·3	287·5	293·8	300·2	306·6
8	313·1	319·6	326·3	333·0	339·7	396·6	353·5	360·4	367·5	374·5	381·7	389·0
9	396·3	403·6	411·1	418·6	426·2	433·8	441·5	449·3	457·1	465·0	473·0	481·1
10	489·2	497·4	505·7	514·0	522·4	530·8	539·4	548·0	556·6	565·4	574·1	583·0
11	592·0	600·9	610·0	619·2	628·4	637·6	647·0	656·4	665·9	675·4	685·0	694·7
12	694·5	714·3	724·2	734·1	744·1	754·1	764·4	774·6	784·9	795·2	805·7	816·2
13	826·7	837·4	848·1	858·9	869·7	880·6	891·6	902·6	913·7	924·9	936·1	947·5
14	958·8	970·3	981·8	993·4	1,005	1,017	1,029	1,040	1,052	1,064	1,076	1,089
15	1,101	1,113	1,125	1,138	1,150	1,163	1,175	1,188	1,201	1,214	1,226	1,239
16	1,252	1,265	1,279	1,292	1,305	1,318	1,332	1,345	1,359	1,373	1,386	1,400
17	1,414	1,428	1,442	1,456	1,470	1,484	1,498	1,513	1,527	1,541	1,556	1,570
18	1,585	1,600	1,615	1,629	1,644	1,659	1,674	1,689	1,705	1,720	1,735	1,751
19	1,766	1,782	1,797	1,813	1,829	1,844	1,860	1,876	1,892	1,908	1,924	1,941
20	1,957	1,973	1,990	2,006	2,023	2,039	2,056	2,073	2,089	2,106	2,123	2,140

TABLE LXI (Sect. 443). LIQUID SURFACE AND CONTENTS OF HORIZONTAL CYLINDRICAL TANKS

PER CENT. DEPTH	CONTENTS PER CENT. OF VOLUME	SURFACE PER CENT. OF CROSS SECTION
5	1·87	43·7
10	5·20	59·9
15	9·41	71·2
20	14·23	80·0
25	19·55	86·7
30	25·23	91·6
35	31·19	95·4
40	37·36	98·1
45	43·64	99·6
50	50·00	100·0
55	56·36	99·6
60	62·64	98·1
65	68·81	95·4
70	74·77	91·6
75	80·45	86·7
80	85·77	80·0
85	90·59	71·2
90	94·80	59·9
95	98·13	43·7
100	100·00	0

TABLE LXIA (Sect. 464). TEMPERATURE LOSS IN °F. FROM 10,000 GALLONS WATER IN CYLINDRICAL TANK 13 ft. DIA. & 12 ft. 6 in. HIGH, COVERED AND HEAVILY LAGGED

TIME OF COOLING		TEMPERATURE DROP OF WATER FROM 210° F.	
HOURS	DAYS	AIR AT 70° F.	AIR AT 50° F.
6		1·7	1·9
12		3·3	3·8
18		4·9	5·6
24	1	6·5	7·5
30		8·1	9·3
36		9·7	11·0
42		11·2	12·8
48	2	12·7	14·5
54		14·2	16·2
60		15·7	17·7
66		17·1	19·6
72	3	18·5	21·7
78		19·9	22·8
84		21·3	24·4
90		22·7	26·0
96	4	24·1	27·5
102		25·4	29·0
108		26·7	30·3
114		28·0	32·0
120	5	29·3	33·5
126		30·6	35·0
132		31·9	36·5
138		33·1	37·8
144	6	34·3	39·3
150		35·5	40·6
156		36·7	42·0
162		37·9	43·3
168	7	39·1	44·6
174		40·2	46·0
180		41·4	47·3
186		42·5	48·6
192	8	43·6	49·9

TABLE LXII (Sect. 563). PROBABLE HEAT CONTENTS OF FUELS

FUEL	HEAT CONTENT BTU	
	SECONDARY (TECHNICAL)	PRIMARY
Anthracite	15,000 per lb.	15,000 per lb.
Hard steam coal	14,000 ,,	14,000 ,,
Good bituminous coal	13,000 ,,	13,000 ,,
Average bituminous coal	12,000 ,,	12,000 ,,
Poor bituminous coal	11,000 ,,	11,000 ,,
Gas works coke	12,500 ,,	16,275 ,,
Coke oven coke	12,500 ,,	15,650 ,,
Wood	5,500 ,,	5,500 ,,
Petrol	20,000 ,,	? 30,000 ,,
Fuel oil	19,000 ,,	? 20,000 ,,
Coal tar oil	16,500 ,,	? 16,500 ,,
Creosote pitch	16,500 ,,	? 16,500 ,,
Coke oven gas	550 per cu. ft.	690* per cu. ft.
Town gas	500 ,,	650* ,,
Water gas	290 ,,	?
Producer gas—hot	160 ,,	180 ,,
,, ,, —cold	130 ,,	180 ,,
Blast furnace gas—cold	100 ,,	100 ,,
Electricity—Best condensing stations ..	3,415 per kWh	10,600† per kWh
,, —Grid average	3,415 ,,	14,000† ,,
,, —Grid peaks	3,415 ,,	30,000† ,,
,, —Back pressure—large good ..	3,415 ,,	5,070 ,,
,, — ,, ,, —small good ..	3,415 ,,	9,600 ,,
,, — ,, ,, —small bad	3,415 ,,	12,000 ,,

*† No distribution loss has been allowed for.

* The gas figures should be increased by about 2 per cent. to cover distribution loss.

† These figures should be increased by about 10 per cent. to cover distribution losses.

TABLE LXIII (Sect. 571). COST TO OWNER AND TO NATION OF HEATING A ROOM

TABLE LXIV (Sect. 571). HEATING A FURNACE

TABLE LXV (Sect. 573). "BOGEY" BOILER EFFICIENCIES

	PER CENT.
Crane boilers	50
Shunting loco boilers	60
Vertical cross tube boilers—small	70
„ „ „ „ —large	78
Lancashire boilers	75
Economic boilers	78
Watertube boilers—small	78
„ „ —large	84
„ „ —very large	90
Sectional boilers—small	70
„ „ —large	75

TABLE LXVI (Sect. 574). PROBABLE BOILER EFFICIENCIES

	LANCASHIRE	ECONOMIC	WATERTUBE
No instruments, dirty, brickwork leaking badly, dampers jammed .. .. .. (A)	40	45	50
Few instruments, clean, brickwork leaking, dampers inaccessible .. .. .. (A)	50	55	60
No instruments, clean, brickwork fair, dampers accessible .. .. (B)	60	65	68
Good instruments, all working, tight brickwork, dampers operated from firing floor .. (B)	65	70	73
Deduct 5 if working day work only (A) Add 5 for economiser (B) Add 8 for economiser			

TABLE LXVII (Sect. 586). APPROXIMATE HEAT REQUIREMENTS FOR BRITISH WINTER VENTILATION

Ultra-lavish ventilation	4 Btu/cu. ft./hour
Lavish ventilation	3 „ „ „
Good ventilation	2 „ „ „
Moderate ventilation	1 „ „ „
Slight ventilation	·5 „ „ „

TABLE LXVIII (Sect. 587). APPROXIMATE BRITISH WINTER HEAT LOSS FROM BUILDINGS

WALLS	BTU/SQ. FT. HOUR	ROOFS	BTU/SQ. FT. HOUR
Single glass	28	..	36
Double glass	15	..	20
Plain brick, 4½-in.	18	Tiles on battens	45
„ „ 4½-in. + fibreboard + air space	8	„ „ boards	30
„ „ 9-in.	14	„ „ „ and felt	11
„ „ 14-in.	11		
„ „ 18-in.	10		
„ „ 23-in.	8		
For plastered walls .. deduct	1		
Plastered brick cavity walls, 11-in.	9	Asphalte on 6-in. hollow tiles	15
„ „ „ „ 16-in.	8	„ „ „ „ „ + 1-in. cork	6
„ „ „ „ 20-in.	7		
Stone, 12-in.	15		
„ 18-in.	12		
„ 24-in.	10		
Concrete, 4-in.	18		
„ 4-in. + fibreboard + air space	8		
„ 6-in.	16	6-in. + Asphalte	17
„ 8-in.	14	„ „ + 1-in. cork	6
„ 10-in.	12	„ „ + fibreboard shuttering	11
		„ „ „ + air space	8
Corrugated iron	34	..	45
„ „ + fibreboard + air space	9	..	10
„ asbestos	32	..	42
„ „ + fibreboard + air space	9	..	10
„ „ on wood tongued and grooved	12		
Plain asbestos	25		
„ „ + fibreboard + air space	8		
Wood tongued and grooved, 1½-in.	12	+ felt	11

FLOORS

Ventilated wood—bare boards	10
„ „ —boards + lino or rubber	9
Concrete—bare or granolithic or tiled	6
„ + wood block	5

TABLE LXIX (Sect. 589). BOGEY SOCIAL WATER HEATING

Washing	
5-in. bath	12,000 Btu/use
Total immersion bath	20,000 „ „
Shower	11,000 „ „
Basin	1,200 „ „
Canteen	
Up to 100 persons (½ gall. at 160° F., ½ gall. at 180° F.)	1,200 Btu/person
100 to 500 persons (⅓ gall. at 160° F., ⅓ gall. at 180° F.)	800 „ „
500 to 1,000 persons (¼ gall. at 160° F., ¼ gall. at 180° F.)	600 „ „

TABLE LXX (Sect. 590). SOME PHYSICAL CONSTANTS

	LATENT HEAT BTU/LB.		SPECIFIC HEAT
	FUSION	VAPORISATION	BTU/LB./° F.
Water	144	971	1·0
Copper	75	—	·1
Iron	45	—	·15
Lead	9·5	—	·03
Ammonia	—	540	·5
Carbon dioxide	—	100	·25
Hydrogen	—	—	3·3
Bottle glass	—	—	·15
Many organic solids	30	—	·25
Many organic liquids	—	125	·5
Most ordinary gases	—	—	·25
Most textiles	—	—	·35
Most other dry substances	—	—	·25

TABLE LXXI (Sect. 683). CORRELATION SIGNIFICANCE

TABLE LXXII (Sect. 686). ERROR RATIO "t"

TABLE LXXIII (Sect. 702). 8-POLE SQUIRREL CAGE MOTORS ON IDENTICAL LOADS (440 V. 50 CYCLES)

HEAVY LINE INDICATES FULL LOAD

LOAD	3 H.P.		5 H.P.		7·5 H.P.		12·5 H.P.		18 H.P.		20 H.P.		25 H.P.	
	STAR	DELTA	STAR	DELTA	STAR	DELTA	STAR	DELTA	STAR	DELTA	STAR	DELTA	STAR	DELTA
TRUE KILOWATTS.														
Motor alone . .	·008	·80	·09	1·83	·24	2·46	·22	2·38	·66	4·99	·91	5·21	1·32	6·53
Motor + Generator	1·28	1·78	1·88	2·91	1·78	3·57	2·27	3·52	2·26	6·03	2·40	6·33	2·47	7·22
M+G+1 H.P.	—	3·35	—	3·41	3·22	4·26	3·00	4·32	3·20	6·32	3·71	6·73	3·35	7·33
M+G+2 H.P.	—	4·84	—	4·28	4·92	4·80	4·11	4·85	3·60	6·83	4·40	7·19	4·11	7·86
M+G+3 H.P.	—	7·29	—	5·28	—	5·42	6·36	5·28	4·87	7·78	5·11	7·73	4·99	8·25
M+G+4 H.P.	—		—	7·92	—	7·61	—	7·31	7·22	9·38	8·20	8·79	7·31	9·71
M+G+5 H.P.	—				—	11·93	—	9·79	—	12·33	—	11·71	11·41	11·72
M+G+6 H.P.	—				—	20·20	—	13·43	—	14·36	—	15·16	—	14·80
POWER FACTOR														
Motor alone . .	·02	·38	·15	·50	·24	·53	·21	·52	·36	·58	·40	·58	·48	·60
Motor + Generator	·56	·60	·65	·66	·65	·66	·71	·66	·66	·66	·70	·66	·65	·65
M+G+1 H.P.	—	·71	—	·69	·73	·70	·73	·71	·725	·67	·75	·68	·71	·66
M+G+2 H.P.	—	·74	—	·73	·74	·73	·75	·75	·75	·69	·77	·70	·75	·68
M+G+3 H.P.	—	·76	—	·77	—	·75	·76	·77	·78	·72	·79	·73	·78	·69
M+G+4 H.P.	—		—	·80	—	·80	—	·80	·79	·77	·81	·76	·80	·75
M+G+5 H.P.						·83	—	·83	—	·81	—	·81	·81	·77
M+G+6 H.P.						·83	—	·84		·82	—	·83	—	·81
MOTOR SPEED														
Motor alone . .	735	745	745	745	745	745	745	745	740	740	745	745	745	745
Motor + Generator	670	730	720	735	735	740	735	745	740	740	740	740	740	745
M+G+1 H.P.	—	715	—	735	720	740	730	740	735	740	735	740	740	745
M+G+2 H.P.	—	700	—	725	660	735	720	740	725	740	735	740	735	745
M+G+3 H.P.	—	650	—	720	—	730	700	740	720	735	730	740	735	740
M+G+4 H.P.			—	710	—	725	—	735	695	735	710	735	720	740
M+G+5 H.P.					—	710	—	730	—	735	—	735	700	740
M+G+6 H.P.					—	665	—	725	—	735	—	735	—	740

TABLE LXXIV (Sect. 725). APPROXIMATE DEW POINTS OF FLUE GASES

FUEL	DEW POINT
Blast furnace gas	55° F. to 60° F.
Producer Gas—Coke	85° F.
Producer Gas—Coal	140° F.
Town Gas	140° F. to 150° F.
Coal—Good Midland	80° F. to 150° F.*
Coal—Bad, Sulphurous	Up to 300° F.

*These figures must not be accepted as authoritative. Apparently good coals sometimes give a dew point of over 250° F.

TABLE LXXV (Sect. 770). HEAT PUMP PERFORMANCE

TABLE LXXVI (Sect. 775). HEAT PUMP PERFORMANCE

TABLE LXXVII (Sect. 793). SPECIFIC HEAT OF ICE

TEMPERATURE ° F.	APPROXIMATE SPECIFIC HEAT
32	·5
− 60	·4
− 165	·3
− 275	·2
− 375	·1
− 400	·05

TABLE LXXVIII (Sect. 795). PHYSICAL PROPERTIES OF LIQUIDS

	ATMOSPHERIC BOILING POINT ° F.	SPECIFIC HEAT BTU/LB.	LATENT HEAT BTU/LB.	TOTAL HEAT OF VAPOUR AT ATMOSPHERIC BOILING POINT BTU/LB.
Water	212	1·0	971	1,151
Acetone	133	·51	223	275
Aniline	356	·51	187	352
Alcohol—Ethyl ..	164	·68	368	458
Alcohol—Methyl ..	148	·61	473	544
Benzene	176	·47	169	237
Carbon Tetrachloride..	170	·20	84	112
Chloroform	142	·24	100	132
Cresol	394	·55	181	380
Glycol	387	·58	344	551
Mercury	675	·033	122	146
Toluene..	231	·53	156	261

TABLE LXXIX (Sect. 797). VISCOSITY OF WATER

TEMPERATURE ° F.	VISCOSITY	
	CENTIPOISES	LB./SEC. FT.
32	1·79	·001203
40	1·54	·001035
60	1·12	·000753
80	·86	·000578
100	·64	·000430
120	·56	·000376
140	·47	·000316
160	·40	·000269
180	·35	·000235
200	·30	·000200
212	·28	·000190

TABLE LXXX (Sect. 800). COMPOSITION OF VARIOUS WATERS

	PARTS PER 100,000				
	BURTON WELL	DUBLIN RIVER	LONDON M.W.B.	LIVERPOOL CORPORATION	IRISH SEA
Total dissolved solids	165·7	31·4	48·8	6·9	3,386·0
Calcium carbonate ..	10·90	20·30	19·40	—	4·8
Calcium sulphate ..	111·24	6·36	16·05	3·14	133·2
Magnesium carbonate	30·44	1·29	—	—	Tr.
Magnesium chloride	—	—	4·16	1·43	—
Magnesium sulphate	—	—	·80	·29	—
Potassium sulphate ..	1·99	—	—	—	—
Sodium chloride ..	5·57	2·61	1·88	1·14	2,643·9
Sodium nitrate ..	2·81	—	5·36	—	—
Others including iron and silica	1·40	·70	1·00	·39	604·1
	RAIN WATER, LAND'S END	RAIN WATER, LONDON	DEW, ROTHAMSTEAD	SNOW, KEMSING	
Total dissolved solids	42·8	2·76	4·87	4·25	
Organic carbon ..	·131	·383	·264	·306*	
Nitrogen	·054	·258	·297	·003?	
Chlorine	21·8	·50	·53	·55	
Hardness	10·0	1·1	1·9	·95	
	PER CENT. BY VOLUME				
	RAIN WATER	LOCH KATRINE WATER			
Nitrogen	1·308	1·731			
Oxygen	·637	·704			
Carbon dioxide ..	·128	·113			

* There were also 3·0 parts of carbon as visible flocs in suspension.

TABLE LXXXI. CONVERSION FACTORS

Length:

1 ft.	=	12·0 in.	1 in.	=	0·083,333	ft.
	=	0·333,333 yard	1 yard	=	3·0	„
	=	0·060,606 rod	1 rod	=	16·5	„
	=	0·015,151, 5 chain	1 chain	=	66·0	„
	=	0·001,515,15 furlong	1 furlong	=	660·0	„
	=	0·001,388,88 cable	1 cable	=	720·0	„
	=	0·000,189,39 mile	1 mile	=	5,280·0	„
	=	0·000,164,47 naut. m.	1 naut. m.	=	6,080·2	„
	=	304,801 μ	1μ	=	0·000,003,28	„
	=	304·801 mm.	1 mm.	=	0·003,281	„
	=	30·4801 cm.	1 cm.	=	0·032,808	„
	=	0·304,801 m.	1 m.	=	3·280,83	„
	=	0·000,304,8 km	1 km.	=	3,280·83	„

Simple approximations : 0·001 in. = 0·025 mm.
0·0004 in. = 0·01 mm

Area:

1 ft.^{2}	=	144·0 in.^{2}	1 in.^{2}	=	·006,944	ft^2.
	=	0·111,111 yard^2	1 yard^2	=	9·0	„
	=	0·003,67 rod^2	1 rod^2	=	272·25	„
	=	0·000,229,6 chain^2	1 chain^2	=	4,356·0	„
	=	0·000,022,96 acre	1 acre	=	43,560·0	„
	=	0·000,000,035,9 mile^2	1 mile^2	=	27,878,400·0	„
	=	92,900·0 mm.^{2}	1 mm.^{2}	=	0·000,010,76	„
	=	929·0 cm.^{2}	1 cm.^{2}	=	0·001,076,4	„
	=	0·0929 m.^{2}	1 m.^{2}	=	10·763,9	„
	=	0·000,009,29 hectare	1 hectare	=	107,639·3	„
	=	0·000,000,092,9 km.^{2}	1 km.^{2}	=	10,763,730·0	„

π	=	3·1415927	=	Circumference of circle of radius = 1.
		4·0	=	Area of outside square.
		2·0	=	Area of inscribed square.
$\sqrt{2}$	=	1·4142	=	Side of inscribed square.
$\sqrt{\pi}$	=	1·7726	=	Side of equal square.
πr^2	=	0·7854d^2	=	Area of circle.
		πd^2	=	Area of sphere.

TABLE LXXXI. CONVERSION FACTORS—*continued*

Volume:

1 cu. ft.	= 1,728·0 cu. in.	1 cu. in.	=	0·000,579	cu. ft.
	= 0·037,04 cu. yd.	1 cu. yd.	=	27·0	,,
	= 6·228,8 Imp. gall.	1 Imp. gall.	=	0·160,54	,,
	= 24·915,2 Imp. qt.	1 Imp. qt.	=	0·040,135	,,
	= 49·830,4 Imp. pint	1 Imp. pint	=	0·020,067,5	,,
	= 199·322 Imp. gill	1 Imp. gill	=	0·005,017	,,
	= 996·610 fluid oz. (Ap.)	1 fluid oz.	=	0·001,003,4	,,
	= 7,972·880 drachm (Ap.)	1 drachm	=	0·000,125,4	,,
	= 478,372·8 minim (Ap.)	1 minim	=	0·000,002,09	,,
	= 3·114,4 peck	1 peck	=	0·321,09	,,
	= 0·778,6 bushel	1 bushel	=	1·284,35	,,
	= 0·028,318 cu. m.	1 cu. m.	=	35·31	,,
	= 0·283,18 hectolitre	1 hectolitre	=	3·531	,,
	= 28·318 litre	1 litre	=	0·035,31	,,
	= 28,318·0 cc.	1 cc.	=	0·000,035,31	,,

1 Imp. gall. = 1·200,91 U.S. gall.
72 Imp. gall. = 1 puncheon
36 Imp. gall. = 1 barrel
1 standard = 165 cu. ft.

$\frac{\pi}{6}d^3 = 0\cdot524d^3$ = volume of sphere

Weight:

1 lb. (avrdp.)	= 0·071,429 stone	1 stone	=	14·0	lb.
	= 0·035,714 quarter	1 quarter	=	28·0	,,
	= 0·008,928,6 cwt.	1 cwt.	=	112·0	,,
	= 0·000,446,43 ton	1 ton	=	2,240·0	,,
	= 0·000,5 short ton	1 short ton	=	2,000·0	,,
	= 16·0 oz. (Av.)	1 oz. (Av.)	=	0·062,5	,,
	= 256·0 dram (Av.)	1 dram (Av.)	=	0·003,906	,,
	= 7,000·0 grains	1 grain	=	0·000,142,86	,,
	= 1·215,28 lb. (Tr. & Ap.)	1 lb. (Tr. & Ap.)	=	0·822,857	,,
	= 291·666 dwt. (Tr.)	1 dwt.	=	0·003,429	,,
	= 350·0 scruple (Ap.)	1 scruple	=	0·002,857	,,
	= 453·592,4 gram	1 gram	=	0·002,204,6	,,
	= 453,592·4 mg.	1 mg.	=	0·000,002,205	,,
	= 0·453,592 kg.	1 kg.	=	2·204,622	,,
	= 0·004,536 quintal	1 quintal	=	220·462,2	,,
	= 0·000,453,6 met. ton	1 met. ton	=	2,204·622	,,

1 carat = 200 mg.

TABLE LXXXI. CONVERSION FACTORS—*continued*

Density :

1 lb./ft.^{3}	=	0·000,578,7 lb./in.^{3}	1 lb./in.^{3}	=	1,728·0	lb./ft.^{3}
	=	27·0 lb./yd.^{3}	1 lb./yd.^{3}	=	0·037,037	,,
	=	0·160,545 lb./gal.	1 lb./gal.	=	6·228,8	,,
	=	0·012,054 ton/yd.^{3}	1 ton/yd.^{3}	=	82·963	,,
	=	0·016,018 met./ ton/m^3	1 met. ton/m^3	=	62·428	,,
	=	16·018,4 kg./m^3	1 kg./m^3	=	0·062,428	,,
	=	0·016,018 gm./cc.	1 gm./cc.	=	62·428	,,

1 ft.^{3} air = ·080,73 lb.
1 ft.^{3} H_2O = 62·288 lb.

Angle :

1 circle	=	360·0 degree	1 degree	=	0·002,78	circle
	=	21,600·0 minutes	1 minute	=	0·000,046,3	,,
	=	1,296,000·0 seconds	1 second	=	0·000,000,77	,,
	=	6·283,185 radians	1 radian	=	0·159,155	,,

Pressure :

1 psi	=	144·0 lb./ft.^{2}	1 lb./ft.^{2}	=	0·006,944	psi
	=	0·000,446,4 ton/in.	1 ton/in.^{2}	=	2,240·0	,,
	=	0·064,28 ton/ft.^{2}	1 ton/ft.^{2}	=	15·556	,,
	=	0·070,31 kg/cm.^{2}	1 kg./cm.^{2}	=	14·22	,,
	=	703·1 kg./m.^{2}	1 kg./m.^{2}	=	0·001,422	,,
	=	2·036 in. hg.	1 in. hg.	=	0·491,2	,,
	=	51·71 mm. hg.	1 mm. hg.	=	0·019,34	,,
	=	0·068,04 atmos.	1 atmos.	=	14·696	,,
	=	27·7 in. H_2O	1 in. H_2O	=	0·036,1	,,
	=	2·309 ft. H_2O	1 ft. H_2O	=	0·433,2	,,
	=	70·37 cm. H_2O	1 cm. H_2O	=	0·014,21	,,

Velocity :

1 ft./sec.	=	60·0 ft./min.	1 ft./min.	=	0·016,667	ft./sec.
	=	0·681,8 m.p.h.	1 m.p.h.	=	1·467	,,
	=	0·011,36 m.p.min.	1 m.p.min.	=	88·0	,,
	=	0·592,09 knot	1 knot	=	1·688,9	,,
	=	30·48 cm./sec.	1 cm./sec.	=	0·032,81	,,
	=	0·304,8 m./sec.	1 m./sec.	=	3·281	,,
	=	18·29 m./min.	1 m./min.	=	0·054,7	,,
	=	0·018,29 km./min.	1 km./min.	=	54·68	,,
	=	1·097 km./hr.	1 km./hr.	=	0·911,3	,,
1 r.p.m.	=	0·016,667 r.p.sec.	1 r.p.sec.	=	60·0	r.p.m.
	=	6·0 deg./sec.	1 deg./sec.	=	0·166,67	,,
	=	0·104,7 radian/sec.	1 radian/sec.	=	9·549	,,

Velocity of light = 186,330 mile/sec.
Velocity of sound = 1,126 ft./sec.

Acceleration due to gravity = g = 32·16 ft./sec.
= 9·81 m./sec.

TABLE LXXXI. CONVERSION FACTORS—*continued*

Flow :

1 gal/sec. =	60·0 gal./min.	1 gal./min.	=	0·016,667	gal./sec.
=	3,600·0 gal./hr.	1 gal./hr.	=	0·000,277,8	„
=	0·160,54 ft.³/sec.	1 ft.³/sec.	=	6·228,8	„
=	9·632,4 ft.³/min.	1 ft.³/min.	=	0·103,813	„
=	577·944 ft.³/hr.	1 ft.³/hr.	=	0·001,730,2	„
=	4·545,63 litre/sec.	1 litre/sec.	=	0·219,999	„
=	272·737,8 litre/min.	1 litre/min.	=	0·003,666	„
=	16,364·268 litre/hr.	1 litre/hr.	=	0·000,061,1	„

Energy :

1 kWh =	1·341 h.p./hr.	1 h.p.-hr.	=	0·745,7	kWh
=	3,415·0 Btu	1 Btu	=	0·000,292,8	„
=	1,897·2 C.H.U.	1 C.H.U.	=	0·000,527,04	„
=	2,655,000·0 ft.-lb.	1 ft.-lb.	=	0·000,000,377	„
=	367,100·0 kg.-m.	1 kg.-m.	=	0·000,002,724	„
=	3,600,000·0 Joule	1 Joule	=	0·000,000,278	„
=	860,445·0 gm.-cal.	1 gm.-cal.	=	0·000,001,162	„

1 ft.-lb. = 13,560,000 erg.

981·2 Dyne = 1 gm.

Power :

1 kW =	1·341 H.P.	1 H.P.	=	0·745,7	kW
=	1·36 Met.H.P.	1 Met.H.P.	=	0·735,5	„
=	737·6 ft.-lb./sec.	1 ft.-lb./sec.	=	0·001,356	„
=	102·0 kg.-m./sec.	1 kg.-m./sec.	=	0·009,807	„
=	1,000·0 Joule/sec.	1 Joule/sec.	=	0·001	„
=	0·239 kg.-cal./sec.	1 kg.-cal./sec.	=	4·183	„
=	0·948,6 Btu/sec.	1 Btu/sec.	=	1·054	„
=	56·916 Btu/min.	1 Btu/min.	=	0·017,567	„
=	3,414·96 Btu/hr.	1 Btu/hr.	=	0·000,292,8	„
=	0·527 chu/sec.	1 chu/sec.	=	1·897,23	„
=	31·62 chu/min.	1 chu/min.	=	0·031,621	„
=	1,897·2 chu/hr.	1 chu/hr.	=	0·000,527	„
=	10,000,000,000 erg./sec.	1 erg./sec.	=	0·000,000,000,1	„

H.P. = heat drop × steam flow/hr. × ·00039

1 boiler H.P. = 34·5 lb./hr. From and At 212° F.

$\sqrt{3}$ = 1·732,050,8

$\frac{1}{\sqrt{3}}$ = 0·577,350,3

TABLE LXXXI. CONVERSION FACTORS—*continued*

Heat Transfer :						
1 Btu/ft.^{2}/hr.	=	0·5556 CHU/ft.^{2}/hr.	1 CHU/ft.^{2}/hr.	=	1·8	Btu/ft.^{2}/hr.
	=	0·009259 CHU/ft.^{2}/min.	1 CHU/ft.^{2}/min.	=	108·0	,,
	=	0·0001543 CHU/ft.^{2}/sec.	1 CHU/ft.^{2}/sec.	=	6480·0	,,
	=	0·27108 cal./cm.^{2}/hr.	1 cal./cm.^{2}/hr.	=	3·687	,,
	=	0·004518 cal./cm.^{2}/min.	1 cal./cm.^{2}/min.	=	221·22	,,
	=	0·00007527 cal./cm.^{2}/sec. (cgs)	1 cal./cm.^{2}/sec.	=	13273·2	,,
	=	2·7108 Kcal./m.^{2}/hr.	1 Kcal./m.^{2}/hr.	=	0·3687	,,
	=	0·004518 Kcal./m.^{2}/min.	1 Kcal./m.^{2}/min.	=	22·12	,,
	=	0·000753 Kcal./m.^{2}/sec.	1 Kcal./m.^{2}/sec.	=	1327·32	,,
	=	0·252 Kcal./ft.^{2}/hr.	1 Kcal./ft.^{2}/hr.	=	3·97	,,
	=	0·0042 Kcal./ft.^{2}/min.	1 Kcal./ft.^{2}/min.	=	238·2	,,
	=	0·00007 Kcal./ft.^{2}/sec.	1 Kcal./ft.^{2}/sec.	=	14292·0	,,
	=	0·0003152 Watt/cm.^{2}	1 Watt/cm.^{2}	=	3173·0	,,
	=	0·00204 Watt/in.^{2}	1 Watt/in.^{2}	=	490·2	,,
	=	0·000394 HP/ft.^{2}	1 HP/ft.	=	2538·0	,,
Heat Transfer Rate :						
1 Btu/ft.^{2}/hr./°F.	=	1 CHU/ft.^{2}/hr./°C.	1 CHU/ft.^{2}/hr./°C.	=	1·0	Btu/ft.^{2}/hr./°F.
	=	0·4878 cal./cm.^{2}/hr./°C.	1 cal./cm.^{2}/hr./°C.	=	2·05	,,
	=	0·00813 cal./cm.^{2}/min./°C.	1 cal./cm.^{2}/min./°C.	=	123·0	,,
	=	0·0001355 cal./cm.^{2}/sec./°C.	1 cal./cm.^{2}/sec./°C.	=	7380·0	,,
	=	4·878 Kcal./m.^{2}/hr./°C.	1 Kcal./m.^{2}/hr./°C.	=	0·205	,,
	=	0·0813 Kcal./m.^{2}/min./°C.	1 Kcal./m.^{2}/min./°C.	=	12·3	,,
	=	0·001355 Kcal./m.^{2}/sec./°C.	1 Kcal./m.^{2}/sec./°C.	=	738·0	,,
	=	0·454 Kcal./ft.^{2}/hr./°C.	1 Kcal./ft.^{2}/hr./°C.	=	2·205	,,
	=	0·00757 Kcal./ft.^{2}/min./°C.	1 Kcal./ft.^{2}/min./°C.	=	132·3	,,
	=	0·000126 Kcal./ft.^{2}/sec./°C	1 Kcal./ft.^{2}/sec./°C.	=	7938·0	,,
	=	0·000568 Watt/cm.^{2}/°C.	1 Watt/cm.^{2}/°C.	=	1761·0	,,
	=	0·00204 Watt/in.^{2}/°F.	1 Watt/in.^{2}/°F.	=	490·2	,,
	=	0·000394 HP/ft.^{2}/°F.	1 HP/ft.^{2}/°F.	=	2545·0	,,

TABLE LXXXII. COMMON LOGARITHMS

(To Base 10)

	0	1	2	3	4	5	6	7	8	9	Proportional Parts 1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	4516	5428	1	3	4	5	6	8	9	10	11

TABLE LXXXII. COMMON LOGARITHMS—*continued*

	0	1	2	3	4	5	6	7	8	9	PROPORTIONAL PARTS 1	2	3	4	5	6	7	8	9
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7

TABLE LXXXII. COMMON LOGARITHMS—*continued*

	0	1	2	3	4	5	6	7	8	9	PP 1	PP 2	PP 3	PP 4	PP 5	PP 6	PP 7	PP 8	PP 9
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5

TABLE LXXXII. COMMON LOGARITHMS—*continued*

	0	1	2	3	4	5	6	7	8	9	PROPORTIONAL PARTS								
											1	2	3	4	5	6	7	8	9
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

Natural or hyperbolic logarithms are calculated to base $e = 2\cdot71828$. Common logarithms can be converted to natural logarithms by multiplying by 2·3026. Natural logarithms can be converted to common logarithms by multiplying by 0·4343.

INDEX

This Index is a section index ; there are no references to pages. Where illustrations are referred to, the reference is printed in *italics*, but the reference is to the Section to which the illustration applies—not to the figure number. Roman figures refer to tables.

Every dash (—) stands for a word in the line preceding it. Where a pair of hyphened words first appear together, one dash is used to represent the hyphened pair. Where a hyphen connects two words after the first appearance of the first word, two dashes are used. Where confusion might arise where two words mean different things when their order is reversed, hyphens have been inserted ; e.g. Gauge, Pressure, *see* Pressure-gauge ; Pressure, Gauge, *see* Gauge-pressure. For alphabetical purposes prepositions, conjunctions and articles are ignored. Cross-indexing, though lavish, is not 100 per cent. complete.

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